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APPLICATION NO.	FILING DATE	FIRST NAMED INVENTOR	ATTORNEY DOCKET NO.	CONFIRMATION NO.
09/802,982	03/12/2001	Stefan Karl	1200.473	6932

7590 07/21/2005

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EXAMINER	
FORD, JOHN K	

ART UNIT	PAPER NUMBER
3753	

DATE MAILED: 07/21/2005

Please find below and/or attached an Office communication concerning this application or proceeding.

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BEFORE THE BOARD OF PATENT APPEALS AND INTERFERENCES

Application Number: 09/802,982
Filing Date: March 12, 2001
Appellant(s): KARL, STEFAN

MAILED

JUL 2 1 2005

Group 3700

Mr. Matthew Stavish
For Appellant

EXAMINER'S ANSWER

This is in response to the appeal brief filed May 18, 2005 appealing from the Office action mailed January 21, 2005.

(1) Real Party in Interest

A statement identifying by name the real party in interest is contained in the brief.

(2) Related Appeals and Interferences

The examiner is not aware of any related appeals, interferences, or judicial proceedings which will directly affect or be directly affected by or have a bearing on the Board's decision in the pending appeal.

(3) Status of Claims

The statement of the status of claims contained in the brief is correct.

(4) Status of Amendments After Final

The appellant's statement of the status of amendments after final rejection contained in the brief is correct.

No amendment after final has been filed.

(5) Summary of Claimed Subject Matter

The summary of claimed subject matter contained in the brief is correct.

(6) Grounds of Rejection to be Reviewed on Appeal

The appellant's statement of the grounds of rejection to be reviewed on appeal is correct.

(7) Claims Appendix

The copy of the appealed claims contained in the Appendix to the brief is correct.

(8) Evidence Relied Upon

The following is a listing of the evidence (e.g., patents, publications, Official Notice, and admitted prior art) relied upon in the rejection of claims under appeal.

JP 10-76837	Noda et al.	3/1998
USP 5,291,941	Enomoto et al	3/1994
USP 5,971,290	Echigoya et al	10/1999
USP 3,910,345	Whalen	10/1975
JP 59-24134	Momose	2/1984
FR 2288278	Patry	5/1976
JP 63-207709	Obara	8/1988
JP 11-34640	Suzuki et al	2/1999

(Note USP 6,047,770 is an equivalent of JP 11-34640)

(9) Grounds of Rejection

The following ground(s) of rejection are applicable to the appealed claims:

Claims 12, 15 and 17-23 are rejected under 35 U.S.C. 112, first paragraph, as failing to comply with the written description requirement. The claim(s) contains subject matter which was not described in the specification in such a way as to reasonably convey to one skilled in the relevant art that the inventor(s), at the time the application was filed, had possession of the claimed invention. To the extent that all of these claims can be read to claim some automatic control system of these valves to produce some intended effect on compressor pressure, there simply is no support in the original specification for any type of automatic control.

No automatic control is disclosed to ensure that this control will occur. Consistent with MPEP 2114, the manner of operating the device does not differentiate an apparatus claim from the prior art. Ex parte Masham, 2 USPQ2d 1647 (BPAI 1987). Regarding the fact that references that show bypasses do not explicitly appear to teach that these bypasses regulate compressor inlet pressure, it is old and well settled law that the motivation of combining references need not be for the same reason as applicant has identified. In re Lintner, 173 USPQ 560 (CCPA 1972) or In re Dillon 16 USPQ2d 1987 (Fed. Cir. 1991).

Claims 2, 8, 9, 11 and 12 are rejected under 35 U.S.C. 112, second paragraph, as being indefinite for failing to particularly point out and distinctly claim the subject matter which applicant regards as the invention.

In claims 2 and 11 it is unclear which of the two claimed evaporators or two claimed condensers of claim 1, applicant is referring to. Make it clear.

Claims 8 and 9 recite an accumulator that doesn't appear to exist in elected Figure 3. Either explain where it is or designate these claims as non-elected in response to this action or amend the claims to describe structure shown in elected Figure 3.

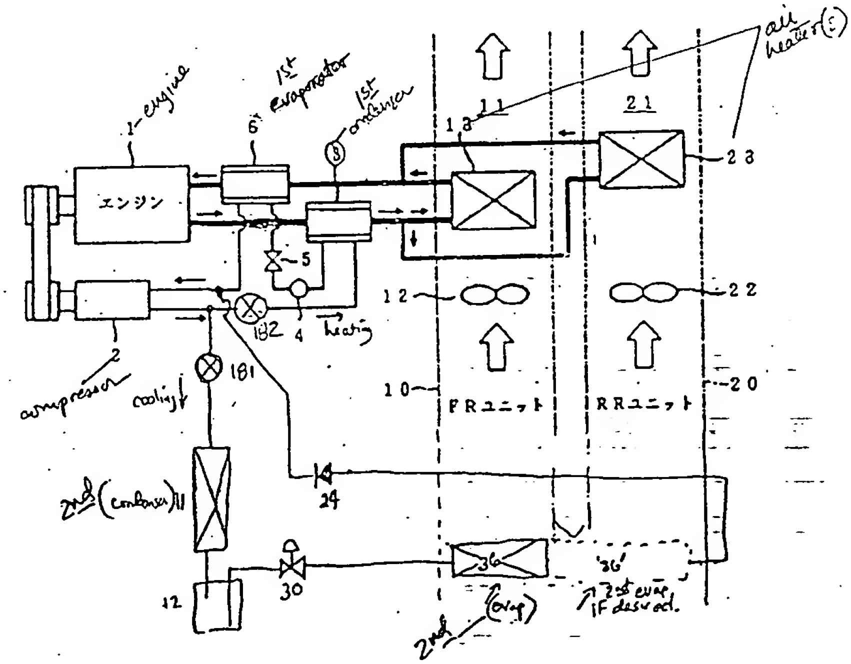
Claim 12 is very vague. A host of not previously claimed or not well-defined structure is found in the second paragraph of that claim including: "first valve system," and the "second valve system." Moreover, the anti-return valve is not upstream of the evaporator, is it?

Claims 1, 2, 10, 11, 13, and 14 are rejected under 35 U.S.C. 103(a) as being unpatentable over the combined teachings of JP 10-76837 and Enomoto, Figure 8, and the description thereof.

JA '837 shows a refrigerant system for beneficially increasing the heating effect of a liquid based heating system. It has no ability to cool the compartment in hot weather. The compressor 2 is known to be the most costly component of automotive air conditioning systems.

Enomoto teaches in Figure 8 a refrigerant based heater circuit (182, 13, 37, 15) and a refrigerant based cooler circuit (181, 11, 12, 30, 36 and 24) connected in parallel across the output and inputs of the compressor 10.

To have added a refrigerant based cooler circuit (as described above) to JA '837 to give the capability of cooling in the summer as well as heating in the winter would have been obvious to one of ordinary skill in climates where air conditioning was needed to preserve occupant comfort. Appropriate valves (181, 182) on the discharge side of the compressor would be necessary to separately activate the heating and cooling systems. The modification is shown below:



Regarding claim 10, see element 24 in the sketch. Regarding claim 11, see elements 5 and 30. Regarding claims 13 and 14, not Enomoto shows an internal combustion engine and discloses as a substitute an electric motor (col.5, last line).

To have used the JA '837/Enomoto system in an electric car or gasoline powered car would have been obvious given the general acceptance of both by the general public.

Claims 8 and 9 are rejected under 35 U.S.C. 103(a) as being unpatentable over the prior art as applied to claim 1 above, and further in view of Echigoya et al.

To have used a conventional suction line accumulator such as disclosed by Echigoya at 66 in the prior art to prevent the compressor from ingesting liquid refrigerant and then breaking would have been obvious to one of ordinary skill in the art.

Claims 12, 15, 17, 18, 19, 20, 21, 22, and 23 are rejected under 35 U.S.C. 103(a) as being unpatentable over the prior art as applied to claim 1 above, and further in view of Whalen or Momose (JP 5-24134) or FR 2288278.

Whalen teaches bypasses 64 and 66 around a chiller 10 and heater 12 controlled by valves 58 and 56. Similarly Momose Fig. 1 shows a heat pump circuit (1, 2, 3 and 4)

and external fluid bypasses in loops 6 and 9. FR '278 (at 9 and 14 with valves 4 and 15, respectively) teaches similar bypasses of the externally circulated fluid.

In Whalen, Momose and FR '278, the bypasses of the external fluids permit more accurate control of its temperature. To have added such coolant bypasses around heat exchangers 6 and 3 of JP 10-76837/Enomoto to permit accurate control of fluid (coolant) temperatures in the coolant loop (i.e. to prevent the engine coolant from getting too cold or too hot) would have been obvious to one of ordinary skill.

Claims 15 and 17 are rejected under 35 U.S.C. 103(a) as being unpatentable over the prior art as applied to claim 1 above, and further in view of JP 63-207,709.

JP '709 teaches air evaporator fluid bypass valve 13, controlling flow to heat exchanger 12 and water valve at 11. To have modified the prior art with such control valves 11 and 13 and a heat exchanger 12 would have been obvious to permit improved control.

Claims 18 and 19 are rejected under 35 U.S.C. 103(a) as being unpatentable over the prior art as applied to claim 1 above, and further in view of JP 11-34640.

JP '640 shows a condenser bypass conduit 47 and a control valve 45 to introduce engine coolant fluid into the condenser heat exchanger 31. To have added such valves and bypasses to the prior art (and, if necessary, outside heat exchanger 42 as shown in JP '640) would have been obvious to more adequately control the heating and to get rid of excess heat.

Claims 12 and 20-23 are rejected under 35 U.S.C. 103(a) as being unpatentable over the prior art as applied to claim 15 above, and further in view of JP 11-34640.

JP '640 shows a condenser bypass conduit 47 and a control valve 45 to introduce engine coolant fluid into the condenser heat exchanger 31. To have added such valves and bypasses to the prior art (and, if necessary, outside heat exchanger 42 as shown in JP '640) would have been obvious to more adequately control the heating and to get rid of excess heat.

(10) Response to Argument

112, first paragraph

Appellant's argument ignores the language in the claim 12, wherein appellant explicitly claims "wherein said first valve system is operatively connected with said second valve **to control** an intake pressure of said compressor" (emphasis supplied). There is no automatic control algorithm or system disclosed in the original specification, claims or drawings to support this limitation. It is submitted that the question is not whether manipulation of the first valve system and second valve could produce some effect on the compressor inlet pressure, but rather how the **control** of intake pressure is accomplished using these valves. Similarly in claim 15, appellant claims "a first valve system **controlling** the amount of heat transferred to said evaporator and thereby

controlling an inlet pressure to said compressor." Again, there is no originally disclosed system to accomplish this desired result. Claims 17 and 21 depend from claim 15 and are rejected for the same reasons applicable to claim 15. In claims 18 and 20, appellant claims that the valve system operatively connects/disconnects the engine cooling loop and the first condenser **based on the requirement/non-requirement for additional heat capacity**. There is no disclosure of any control system in the original specification, let alone one that can figure out whether or not additional heat capacity is required. It is submitted that this type of automatic control would require a plurality of undisclosed sensors connected to an undisclosed control system to perform this function. Appellant has not disclosed any control system whatsoever. Claim 19 depends from claim 18 and is rejected for the same reasons applicable to claim 18. Claim 22 claims that the "second valve system is adapted to **control the loading** of the compressor." Again, controlling compressor loading requires some sort of control system, which appellant has not disclosed. Finally in claim 23, appellant claims "wherein said first and second valve systems **control an intake pressure** of said compressor." Again, undisclosed sensors and an undisclosed control system would be required to perform this function.

Contrary to the argument set forth on page 9 of the Brief, clear control structures are claimed that appellant has neither disclosed nor submitted any evidence to show that they are known. Appellant calls the Board's attention to page 5, lines 9-17 of the specification. It is submitted that that does nothing to help Appellant out of his

quandary. A valve, per se, is not a "control means" sufficient to perform the functions claimed in the claims as the Board should be well aware.

112 second paragraph

Claims 2 and 11 should be amended to refer to either the first or second evaporator and the first and second condenser since applicant has set forth those four separate elements with unique names, as Appellant recognizes in the Brief. As well the heat pump loop and the air conditioning branch share at least the compressor 18, valve 48 and evaporator 16 in common in that no matter which leg of the refrigeration piping is selected by valve 48 (i.e. the heat pump loop or the air conditioning branch), refrigerant flow will occur through these elements. It is not as simple as Appellant makes it appear.

In claims 8 and 9, the refrigerant accumulator is recited as a separate element (apart from the evaporator) as is shown at 44 in non-elected Figure 1. Now Appellant asserts that the inside of the evaporator is an accumulator. In elected Figure 3, there is only an evaporator chamber shown and disclosed at 16. If there was a separate accumulator in the evaporator chamber, it isn't shown, appellant's argument would be persuasive. The language of the claim does not describe the structure shown in elected Figure 3. Instead, the evaporator (chamber) 16 functions as an accumulator, there is no separate accumulator apart from the evaporator. To the extent that applicant claims

two disparate devices, that is in reality (in Figure 3) only one device that performs two functions (evaporation and accumulation), it is not descriptive.

Claim 12 is very vague. The first valve system and second valve system are claimed in the penultimate paragraph of this claim. Int is unclear which of a multiplicity of disclosed valves is being claimed. It is possible that two <u>or more</u> of the following valves could be being claimed (valve 27, 28, 30, 31). It is simply unclear both from a reading of the claim in light of the disclosure as well as from Appellant's response precisely which of these valves is being claimed. Are all four being claimed, three or just two? And, if less than four are being claimed, which ones are they? The claim is simply unclear. As well, the last paragraph of claim 12, claims that the first valve system is connected to the "second valve" (note that this term has no precise antecedent), which is presumed to be one of the valves in the second valve system. Again Appellant's Brief contains no answers to these questions.

103 rejection Noda (JP 10-76837) in view of Enomoto (Figure 8)

Appellant here (Brief page 11) , clearly understands the nature of the rejection echoing the Examiner's statement that it would have been obvious to have added a refrigerant based air –conditioning system to Noda so that the occupants of the vehicle could enjoy the enhanced heating performance of Noda, in cold weather, as well as cooling (from a conventional refrigerant loop) in the summer. Most people in hot areas,

such as in the Washington DC metro area, own automobiles that have a heater for the winter and an air conditioner for the summer, as no doubt counsel and the members of the Board are well aware.

After saying that, Appellant attacks the Examiner's rejection by stating "JP '837 [Noda] gives no indication that adding an air conditioning (a/c) circuit of Enomoto would be at all desirable or would improve the functionality of its heating system." Regarding the "improved functionality" argument, the Examiner agrees that the presence of the added air conditioning loop would not change the heating function because there are two separate systems that share the same compressor but function alternatively (i.e. either the heating system is on or the cooling system is on). Appellant's device is no different in this regard – what Appellant has done, it is submitted, has made an irrelevant observation that applies with equal measure to his own system or the references as combined by the Examiner.

Noda, it is submitted, doesn't discuss air conditioning because his invention doesn't concern air conditioning. To put it another way, if Calsonic Corporation, the assignee of Noda, were to come to the United States and market a vehicle heating system with the improved heater disclosed there, is the United States Patent Office going to stop Calsonic from adding a conventional refrigeration loop to the heating system, which shares the compressor (the most expensive component in a conventional refrigeration system) as taught by Enomoto (Figure 8), to their system (so that it will be marketable to the vast majority of automobile buyers), merely because Noda didn't mention cooling in his disclosure? The answer is no.

Appellant asks why would Noda and Enomoto be combined when Enomoto already has heating and cooling circuits (Brief, page 12, lines 14-16). The answer to that question is extremely easy. One would have added the improved heating circuit of Noda to Enomoto to obtain the advantage of that improved heating performance in the winter. Would this modification change the cooling performance – no it wouldn't, because the two systems function independently of one another (just like Appellant's disclosed system).

Appellant at the bottom of page 12 of the Brief suggests that proper motivation must come from the references themselves. Appellant is simply misinformed. See In re Dillon, 16 USPQ2d 1897 (Fed. Cir. 1990) and In re Lintner, 173 USPQ 560 (CCPA 1972).

On page 13 of the Brief Appellant concedes that the Examiner's rejection is sound, states that the "combination would not achieve the presently claimed invention." The combination shown by the examiner pictorially (reproduced on page 13 of the Brief) clearly meets the limitations of the enumerated claims, Appellant's remarks notwithstanding. In fact, Appellant offers nothing of substance to back up this assertion. For example, on page 14 of the Brief he makes his own sketch and argues that that is how one would combine the references. This is unconvincing. <u>The sketch on page 14 of the Brief has no ability to cool.</u> To put it another way, if Calsonic Corporation, the assignee of Noda, were to come to the United States and market a vehicle heating system with the improved heater disclosed there, is the United States Patent Office going to stop Calsonic from adding a conventional refrigeration loop to the heating

system, which shares the compressor (the most expensive component in a conventional refrigeration system) as taught by Enomoto (Figure 8), to their system (so that it will be marketable to the vast majority of automobile buyers), merely because Noda didn't mention cooling in his disclosure? The answer is no. A heating only system for an automobile is of little economic value in the United States because there is almost no consumer demand for them.

On page 15 of the Brief, Appellant challenges the rejection alleging that the Examiner has no switching means in his sketch. Appellant has simply chosen to ignore the presence of valves 181 and 182 and their operation (disclosed in Enomoto).

On page 16 of the Brief, Appellant alleges that the Examiner hasn't shown him that sketch. The answer is very simple. In claims 1, 2, 10, 11, 13 and 14 (the claims enumerated in this rejection), Appellant has not claimed what he has sketched on page 16 of the Brief, Appellant's erroneous statements to the contrary notwithstanding. Likewise the "modular argument" (Brief, page 16) as to the enumerated claims does not claim a module of any sort. Furthermore, there is no hindsight here because the motivation relied upon (the desirability of having a system for both heating and cooling a vehicle to make occupants comfortable in a climate like that of Washington, DC) is not disclosed by Appellant. Moreover, these combined automotive heating and cooling systems have been around since the 1940's – long before Appellant began inventing.

On page 17 of the Brief, Appellant attacks valves 181 and 182 of Enomoto and their function arguing, without any logic that the Examiner can follow, that they would have to be an automated three way valve. Valves 181 and 182 of Enomoto function

identically to Appellant's valve 48. Moreover, in col. 6, lines 1 and 2, Enomoto explicitly states that valves 181 and 182 could be replaced by a three-way valve (which would make it identical to Appellant's disclosure, in so far as the switching valve is concerned). Note however that none of claims 1, 2, 10, 11, 13 and 14 require a three-way valve. Finally, Appellant attacks Enomoto as not showing the improved heating system of Noda. Attacking references singly in the context of 35 USC 103, arguing that one reference doesn't show what the other does and vice versa, is not germane to what the references as combined fairly teach.

Regarding claim 2 (Brief page 17, bottom), evaporator 6 is upstream of the engine as evidenced by the arrow shown between engine 1 and evaporator 6. Apparently Appellant doesn't understand his own claim language or his own disclosure. Appellant's own evaporator 16 cools the engine coolant in heat exchanger 17 and heat exchanger 17 is located upstream (with respect to the engine coolant flow) of the engine 2. Noda is simply no different, Appellant's comments to the contrary notwithstanding.

Regarding claims 11, 10, 13 and 14 (Brief, page 18) Appellant has stated no separate arguments and accordingly the Examiner should be affirmed on these claims.

Regarding claims 8 and 9, (Brief, page 18), Appellant argues that the accumulator 66 located immediately upstream of the compressor in Echigoya serves only the cooling loop. This is false. Refrigerant passes through the accumulator 66 when the system is in the heating mode and the cooling mode (see Table 1, col. 5, lines 18 –30) as anyone of ordinary skill in the art can see by tracing the path though the

system. It would be improper to protect the compressor from damage by ingesting liquid refrigerant only in the cooling mode (and not in the heating mode) using an accumulator. No reasonable designer of refrigeration equipment would do such a thing.

Regarding claim 12 (Brief, page 19), Noda clearly teaches a module (i.e. an integrated structure in Figures 3 and 4, described in the translation in paragraphs 0037-0039), wherein all of the components that don't need to be located anywhere in particular are integrated together. The exceptions are the compressor (generally mounted on the engine accessory drive system), the outdoor air condenser (generally mounted on the front of the radiator) and the indoor evaporator (generally mounted in the dashboard). It is submitted that Noda teaches putting everything else in a module that would obviously advantageously facilitate easy replacement.

Whalen shows the valve systems claimed notwithstanding Appellant's assertions to the contrary. The fact that Whalen is a building and heating and cooling system versus one for a vehicle is in the Examiner's mind irrelevant. The two technologies are interrelated to the point that in the Examiner's search class, the classification schedule fails at many points to distinguish between the two.

Momose shows the valve systems claimed notwithstanding Appellant's assertions to the contrary. The fact that Momose heats and cools different fluids from those disclosed by Appellant is, in the Examiner's mind, irrelevant. The two

technologies are interrelated to the point that in the Examiner's search class, the classification schedule fails at many points to distinguish between the two.

FR '278, cited to the Examiner by Appellant as relevant, does disclose the valve system for the fluid which is thermally conditioned by the refrigerant that permits to either go through the heat exchanger or bypass around it, as do Momose and Whalen. Appellant has no traversed the Examiner's reason for adding such bypasses (i.e. to prevent overheating or overcooling of the associated heat exchanger).

Regarding claim 15 (Brief, page 20), Appellant here argues that the references don't teach controlling the compressor inlet pressure. The Examiner has relied on the best art to teach the valve systems claimed. The Examiner has looked but has not found the control of compressor inlet pressure using valve systems such as disclosed by Appellant, however, Appellant has not disclosed any system for doing what is claimed (i.e. control of compressor inlet pressure using valve systems such as disclosed by Appellant). See the Examiner's rejections under 35 USC 112, first paragraph, above.

Regarding claims 17-19 (Brief, pages 21-22), the claims do not state the valves have to be separate. In these references the bypass valves are three-way types that have two valves built into one unit, one of which controls the flow through the heat exchanger and one of which controls the flow through the bypass.

Regarding claim 20 (Brief, page 22), Appellant here argues that the references don't teach controlling valves responsive to the requirement for additional heat capacity. The Examiner has relied on the best art to teach the valve systems claimed. The Examiner has looked but has not found the control of valve systems based on a need for additional heat capacity such as claimed by Appellant, however, Appellant has not disclosed any system for doing what is claimed. See the Examiner's rejections under 35 USC 112, first paragraph, above.

Regarding claim 21 (Brief, pages 23), the claims do not state the valves have to be separate. In these references the bypass valves are three-way types that have two valves built into one unit, one of which controls the flow through the heat exchanger and one of which controls the flow through the bypass.

Regarding claim 22 (Brief, page 23), Appellant here argues that the references don't teach controlling the compressor loading. The Examiner has relied on the best art to teach the valve systems claimed. The Examiner has looked but has not found the control of compressor loading using valve systems such as disclosed by Appellant, however, Appellant has not disclosed any system for doing what is claimed (i.e. control of compressor loading using valve systems such as disclosed by Appellant). See the Examiner's rejections under 35 USC 112, first paragraph, above.

Regarding claim 23 (Brief, page 23), Appellant here argues that the references don't teach controlling the compressor inlet pressure. The Examiner has relied on the best art to teach the valve systems claimed. The Examiner has looked but has not found the control of compressor inlet pressure using valve systems such as disclosed by Appellant, however, Appellant has not disclosed any system for doing what is claimed (i.e. control of compressor inlet pressure using valve systems such as disclosed by Appellant). See the Examiner's rejections under 35 USC 112, first paragraph, above.

Regarding claim 15 (Brief, page 25), Appellant here argues that the references don't teach controlling the compressor inlet pressure. The Examiner has relied on the best art to teach the valve systems claimed. The Examiner has looked but has not found the control of compressor inlet pressure using valve systems such as disclosed by Appellant, however, Appellant has not disclosed any system for doing what is claimed (i.e. control of compressor inlet pressure using valve systems such as disclosed by Appellant). See the Examiner's rejections under 35 USC 112, first paragraph, above.

Regarding claims 17-19 (Brief, pages 25-26), the claims do not state the valves have to be separate. In these references the bypass valves are three-way types that have two valves built into one unit, one of which controls the flow through the heat exchanger and one of which controls the flow through the bypass.

Regarding claims 12 and 20-23 (Brief, pages 26-28), Appellant admits that JP 11-34640 shows what it is relied upon by the Examiner to show, but argues that it doesn't show a module (that is disclosed by Noda, Figures 3 and 4) and it doesn't show switching valves (that are disclosed by Enomoto at 181 and 182). Contrary to Appellant's characterization, conduit 47 bypasses the condenser 31, which corresponds to the condenser claimed by Appellant – not condenser 25 of JP 11-34640.

Finally, as to the final allegation of hindsight reconstruction (Brief page 28), Appellant has failed to provide even one showing or challenge to the logic provided by the Examiner for making the proposed combinations. None of those reasons are in anyway difficult to understand to one of ordinary skill in the art and none of them were lifted from Appellant's own teachings in the specification.

(11) Related Proceeding(s) Appendix

No decision rendered by a court or the Board is identified by the examiner in the Related Appeals and Interferences section of this examiner's answer.

The examiner requests the opportunity to present arguments at the oral hearing, if an oral hearing is requested by the Appellant.

For the above reasons, it is believed that the rejections should be sustained.

Respectfully submitted,

John Ford

John K. Ford
Primary Examiner

Conferees:

Mr. Gene Mancene SPE 3753

Ms. Lil Ciric Primary Examiner 3753

LJILJANA CIRIC
PRIMARY EXAMINER

Gene Mancene
Supervisory Patent Examiner
Group 3700

Copy of JP 63-207709

with translation

for Appellant.

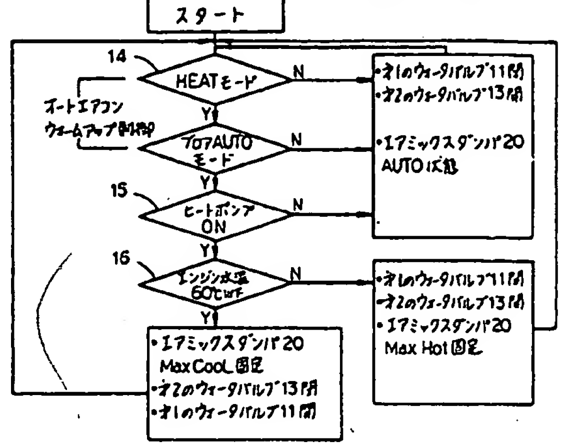
with translation

7

9………エンジン
10a………ヒータ配管路
11………第1のウォータバルブ
12………ヒータコア
13………第2のウォータバルブ
14………モードセンサ
15………ヒートポンプセンサ
16………水温センサ

出願人　　トヨタ自動車株式会社

第 3 図



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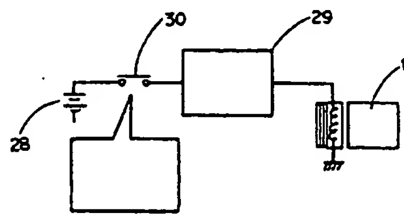
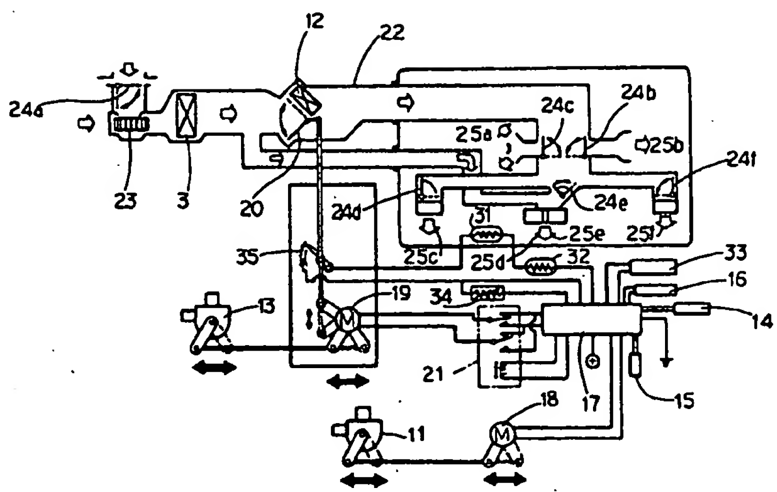


第 1 図

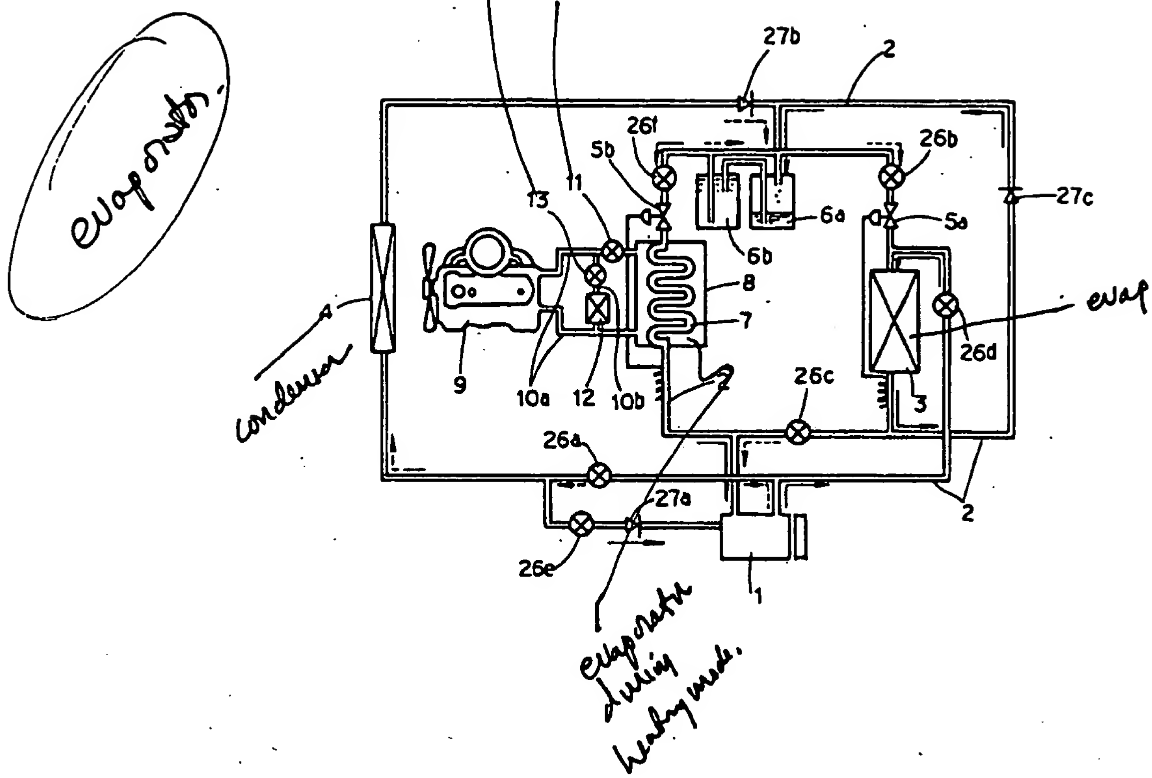


1…コンプレッサ
2…冷媒配管
3…エバポレータ
4…コンデンサ
8…熱交換器
9…エンジン
10a…ヒータ配管路
11…第1のウォータバルブ
12…ヒータコア
13…第2のウォータバルブ
14…モードセンサ
15…ヒートポンプセンサ
16…水温センサ

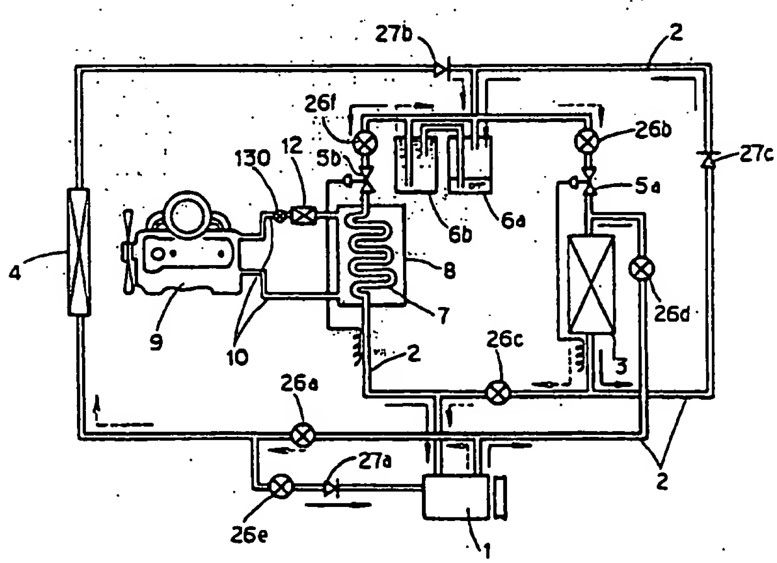
bypass water valve

1st water valve

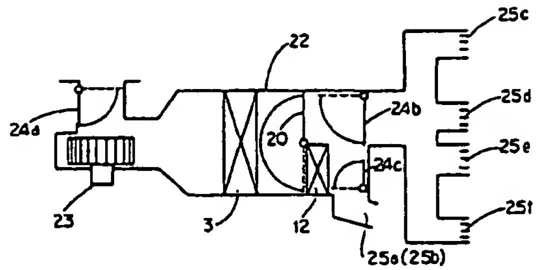
第 2 図



第 5 図



第 6 図



⑲日本国特許庁（JP）　⑪特許出願公開

⑫公開特許公報（A）　昭63-207709

⑤Int.Cl.⁴	識別記号	庁内整理番号	㊸公開　昭和63年(1988)8月29日
B 60 H 1/00	101	C-7153-3L	
F 25 B 27/02		7501-3L	

審査請求　未請求　発明の数 1　（全7頁）

⑭発明の名称　車両用空調装置

㉑特　願　昭62-39649

㉒出　願　昭62(1987)2月23日

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明　細　書

1．発明の名称

車両用空調装置

2．特許請求の範囲

(1)コンプレッサ、コンデンサ、エバポレータを冷媒配管により連結し、その冷媒配管にエンジンの熱源により前記冷媒配管を暖めるヒートポンプ用の熱交換器が備えてあり、冷媒経路を切換えることにより、冷房及び暖房運転に切換自在としたオートエアコン制御の車両用空調装置において、エンジンとヒートポンプ用の熱交換器との間のヒータ配管路に、エンジンからヒートポンプ用の熱交換器への温水流量をコントロールする第1のウォータバルブと、ヒータコアと、ヒータコア用の第2のウォータバルブを設け、ヒートポンプ側の運転情報をモードセンサ、ヒートポンプセンサ、水温センサで入手して、前記第1及び第2のウォータバルブを開閉制御し、ヒートポンプ用の熱交換器及びヒータコアへの温水流量を調節することを特徴とする車両用空調装置。

3．発明の詳細な説明

（産業上の利用分野）

この発明は、冷媒経路を切換えることにより冷房及び暖房運転に切換自在とする車両用空調装置に関するものである。

（従来の技術）

従来のヒートポンプ構造付の車両用空調装置は、第5図及び第6図に示すように、冷凍装置で利用できなかった低い温度の熱源を高温度にして暖房として利用するもので、第5図図示のコンプレッサ1とエバポレータ3とコンデンサ4を冷媒配管2により連結するとともに、前記冷媒配管2からレシーバタンク6a、6bを介して、冷媒配管2をジグザグ状に形成した蛇行成形部7に連結してあり、この冷媒配管2の蛇行成形部7には熱交換器8が覆って形成され、この熱交換器8にはエンジン9の冷却水をウォータバルブ130を備えたヒータ配管路10により循環するようになっている。

そして、暖房運転時には、第5図図示の電磁バ

ルブ26a、26b、26cをOFF(閉)とし、電磁バルブ26d、26e、26fをON(開)として、冷媒配管2内の冷媒液は第5図図中の実線矢印方向に流れる。

一方、前記熱交換器8には、ヒータ配管路10に備えてあるウォータバルブ130を開くことにより、エンジン9からの高温の冷却水がヒータコア12を通って前記熱交換器8に流れ込み、熱交換器8内の冷媒配管2の蛇行成形部7を通過する冷媒液は、エンジン9の高温冷却水から吸熱される。

このように吸熱された冷媒液を、さらにコンプレッサ1により高温に圧縮し、この高温冷媒液が開路される電磁バルブ26dを通ってエバポレータ3に送られ凝縮して熱を吐出する。

第6図は、空調ケース22には送風機23、エバポレータ3、ヒータコア12が配設されており、各ダンパ20及び24a、24b、24cの開閉によって、送風機23からの風を室内への各吹出口25a～25fに供給する空調装置の概略構成が示されているが、このエバポレータ3に供給される前述の高温冷媒液が凝縮して熱を吐出するため、ヒータコア12に加えてエバポレータ3も高温となり、送風機23からの風は暖められて暖房運転となる。

また、冷房運転時には、第5図図示の電磁バルブ26a、26b、26cをON(開)とし、電磁バルブ26d、26e、26fをOFF(閉)として冷媒配管2内の冷媒液はコンデンサ4を通り、第5図図中の破線矢印方向に通常の冷房サイクルのように流れる。

このとき、前記熱交換器8に通じるヒータ配管路10のウォータバルブ130は閉じられていて、熱交換器8内にエンジン9の高温冷却水を流さないようにしてある。

このように、第5図に示すヒートポンプ構造付の車両用空調装置は、クーラに使用した冷凍装置で利用できなかった低い温度の熱源を、クーラサイクルの冷媒液の回路を変えて高温度にしてヒータに利用しようとするものである。

(発明が解決しようとする問題点)

しかし、第5図の場合、ヒータコア12とヒートポンプ用の熱交換器8を同時使用した場合、エンジン水温が低い時に、ヒータコア12での吸熱によりヒートポンプ用の熱交換器8の即効性が低下することがある。

また、ヒータコア12をMaxCool付近で使用する場合、ヒートポンプ用の熱交換器8への通水量が少なくなり、冷風が出たり負圧運転でコンプレッサ故障の原因になることがある。

そこで、このような問題点を解決するため、この考案は、エンジン水温が低い時のヒータコアによる吸熱を防止する機能を工夫することにより、ヒートポンプ用の熱交換の即効性を向上させることにある。

(問題点を解決するための手段)

そのため、この発明は上述の問題点を、エンジンとヒートポンプ用の熱交換器との間のヒータ配管路に、ヒートポンプ用の熱交換器への温水流量をコントロールする第1のウォータバルブと、ヒータコア用の第2のウォータバルブを設けて、これらの各バルブを開閉制御することにより解決しようとするものである。

さらに詳しくは、第1図及び第2図の符号を付して説明すると、コンプレッサ1、コンデンサ4、エバポレータ3を冷媒配管2により連結し、その冷媒配管2にエンジン9の熱源により前記冷媒配管2を暖めるヒートポンプ用の熱交換器8が備えており、冷媒経路を切換えることにより、冷房及び暖房運転に切換自在としたオートエアコン制御の車両用空調装置において、エンジン9とヒートポンプ用の熱交換器8との間のヒータ配管路10aに、エンジン9からヒートポンプ用の熱交換器8への温水流量をコントロールする第1のウォータバルブ11と、ヒータコア12と、ヒータコア12用の第2のウォータバルブ13を設け、ヒートポンプ側の運転情報をモードセンサ14、ヒートポンプセンサ15、水温センサ16で入手して、前記第1及び第2のウォータバルブ11、13を開閉制御し、ヒートポンプ用の熱交換器8及びヒ

ータコア12への温水流量を調節するようにしたものである。

〔作用〕

上述の手段によれば、オートエアコンの制御中において、ヒートポンプ側の運転情報をモードセンサ14、ヒートポンプセンサ15、水温センサ16により入手して、ヒートポンプ用の熱交換器8への温水流量を第1のウォータバルブ11の開閉によりコントロールし、また、ヒータコア12の温水流量を第2のウォータバルブ13の開閉によりコントロールし、エンジン水温が低い時のヒートポンプ用の熱交換器8の即効性を向上させる。

〔実施例〕

以下、添付図面に基づいて、この発明の実施例を説明する。

第1図から第4図までの図面は、この発明の実施例を示しており、ヒートポンプ構造付のオートエアコン制御の車両用空調装置は、第2図図示のコンプレッサ1と、このコンプレッサ1に冷媒配管2を介して連結されるエバポレータ3と、コンプレッサ4が前記冷媒配管2により連結されており、前記エバポレータ3にはその冷媒配管2の管路に第1のエクスパンションバルブ5aが連結されている。

また、前記冷媒配管2からレシーバタンク6a、6b及び第2のエクスパンションバルブ5bを介して、冷媒配管2をジグザグ状に形成した蛇行成形部7に連結してあり、この冷媒配管2の蛇行成形部7にはヒートポンプ用の熱交換器8が覆って形成され、エンジン9と前記熱交換器8との間のヒータ配管路10aには第1のウォータバルブ11を設けて、エンジン9から熱交換器8への温水流量をコントロールする前記第1のウォータバルブ11を開くことにより、エンジン9の温水（冷却水）を前記熱交換器8に循環して、前記冷媒配管2の蛇行成形部7中の冷媒を暖めるようになっており、また、前記ヒータ配管路10aには、さらにバイパス路10bを設けて、このバイパス路10bにヒータコア12を設け、このヒータコア12と前記ヒータ配管路10aの第1のウォータバルブ11との間のバイパス路10bに、ヒータコア12用の第2のウォータバルブ13が設けてある。

一方、第1図から分かるように、オートエアコンの制御中において、ヒートポンプ側の運転情報を入手するセンサとして、モードを判別するモードセンサ14と、ヒートポンプのスイッチのON－OFFを判別するヒートポンプセンサ15と、ヒートポンプ用の熱交換器8内の温水温度を判別する水温センサ16が設けてあり、これらの各センサ14、15、16から入手されるヒートポンプ側の運転情報は、すべてコンピュータ17へ入力信号として与え、第1及び第2のサーボモータ18、19を駆動させるようになっている。

前記第1のサーボモータ18は前記第1のウォータバルブ11（熱交換器8への温水流量コントロール）の開閉を制御し、前記第2のサーボモータ19は、前記第2のウォータバルブ13（ヒータコア12への温水流量コントロール）の開閉を制御するとともに、エアミックスダンパ20を開（HOT（ホット））あるいは閉（COOL（クール））に選択的に開閉するようになっている。

なお、前記コンピュータ17と第2のサーボモータ19の間には切替リレー21を設けて、第2のサーボモータ19の回転を切換えるようになっている。

また、第1図図中の空調ケース22には送風機23、エバポレータ3、ヒータコア12が配設されており、各ダンパ20及び24a、24b、24c、24d、24e、24fの開閉によって、送風機23からの風を室内への各吹出口25a～25fに供給する空調装置の概略構成が示されている。

そして、暖房運転時には、第2図図示の電磁バルブ26a、26b、26cをOFF（閉）とし、電磁バルブ26d、26e、26fをON（開）として、冷媒配管2内の冷媒液は第2図図中の実線矢印方向に流れ、空調装置も暖房運転状態で作動するので、ヒートポンプ用の熱交換器8で吸熱された冷媒液は、さらにコンプレッサ1により高

温に圧縮され、この高温冷媒液が開示されている電磁バルブ26dを通ってエバポレータ3に送られ、凝縮して熱を吐出する。

このように、エバポレータ3に供給される前述の高温冷媒液が凝縮して熱を吐出するため、ヒータコア12に加えてエバポレータ3も高温となり、第1図図示の送風機23からの風は暖められて除湿暖房運転となる。

また、冷房運転時には、第2図図示の電磁バルブ26a、26b、26cをON（開）とし、電磁バルブ26d、26e、26fをOFF（閉）としてその冷媒経路を切換え、冷媒配管2内の冷媒液はコンデンサ4を通って、第2図図中の破線矢印方向に通常の冷房サイクルとして流れる。

このとき、前記熱交換器8に通じるヒータ配管路10aの第1のウォータバルブ11は閉じられていて、熱交換器8内にエンジン9の高温冷却水が流れないようにしてあるから、冷媒液はエバポレータ3に流れ冷房運転となるのである。

なお、第2図図中の符号27a、27b、27cは逆止弁である。

第4図はコンプレッサ1と電源28との間の結線回路に、感温センサ29とコンプレッサスイッチ30を設けたもので、暖房運転時のコンプレッサ1の制御については、エバポレータ3の出口温度制御と、コンプレッサ1の吐出圧力制御であり、エバポレータ3の感温センサ29は、その出口温度が80℃以上でコンプレッサ1をOFFさせ、70℃以下でONさせるとともに、コンプレッサスイッチ30は、コンプレッサ1の吐出圧力が20kg／cm以上でコンプレッサ1をOFFさせ、13、5kg／cm以下でONさせるのである。

なお、第1図図中の符号31は内気センサ、符号32は外気センサ、符号33は日射センサ、符号34は温度設定レバー、符号35はブロア切替スイッチである。

第3図は、この発明の実施例をフローチャートで示したものであり、空調装置のオートエアコン制御の中でウォームアップ制御時に、モードセンサ14及びヒートポンプセンサ15の信号とエンジン9の水温センサ16の信号を受けて、ヒータコア12のエアミックスダンパ20と第2のウォータバルブ13を、切替リレー21を作動させて開閉する。

また、ヒートポンプ用の熱交換器8の第1のウォータバルブ11も同時に開閉させるのである。

（発明の効果）

この発明は上述のように、エンジンとヒートポンプ用の熱交換器との間のヒータ配管路に、ヒートポンプ用の熱交換器への温水流量をコントロールする第1のウォータバルブと、ヒータコア用の第2のウォータバルブを設けて、これらの各バルブを、モードセンサ、ヒートポンプセンサ及び水温センサにより開閉コントロールして、ヒータコアとヒートポンプ用の熱交換器の通水経路を変更するようにしたため、エンジン水温が低い時のヒータコアによる吸熱を防止して、ヒートポンプ用の熱交換の即効性が向上されるとともに、冷風吹き出しと負圧運転でのコンプレッサ故障の防止が図られる。

また、ヒートポンプ使用時はヒータコアをMax Cool（マックスクール）で使う必要があるが、この発明はオートエアコンとの組合せによって、その煩わしさが解消される。

4．図面の簡単な説明

第1図から第4図までの図面は、この発明の実施例を示しており、第1図は空調装置の制御系の構成図、

第2図は冷媒サイクルの概略構成図、

第3図はこの発明要旨のフローチャート、

第4図はセンサ及びスイッチを介して、コンプレッサを電源に接続する結線図、

第5図及び第6図は従来例を示しており、第5図は第2図相当の構成図、

第6図は空調装置の概略構成図である。

1………コンプレッサ
2………冷媒配管
3………エバポレータ
4………コンデンサ
8………熱交換器

PAT-NO: JP363207709A

DOCUMENT-IDENTIFIER: JP 63207709 A

TITLE: AIR CONDITIONER FOR VEHICLE

PUBN-DATE: August 29, 1988

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APPL-NO: JP62039649

APPL-DATE: February 23, 1987

INT-CL (IPC): B60H001/00, F25B027/02

US-CL-CURRENT: 237/12.3R

ABSTRACT:

PURPOSE: To enhance immediate effectiveness of heat-exchange at the time of cooling an engine by providing a water valve for controlling flow quantity to a heat exchanger as well as a water valve for a heater core, in a heater pipeline for transmitting engine cooling water to the heat exchanger in a refrigerant pipeline.

CONSTITUTION: A compressor 1, an evaporator 3 and a condenser 4 are connected to each other by means of refrigerant pipelines 2, and the refrigerant pipelines 2 are connected to a meandering forming part 7 passing through a heat exchanger 8 via receiver tanks 6a, 6b. In the heat exchanger 8, the cooling water for an engine 9 is circulated by means of a heater pipeline having a water valve. In such an air conditioner with a heat pump structure, a first water valve 11 is provided in a heater pipeline 10a between the engine 9 and the heat exchanger 8. Meanwhile, a bypass line 10b is disposed in the heater pipeline 10a, and then, a heater core 12 and a second water valve 13 are provided in the bypass line 10b.

PTO 05-4659 Patent No. Sho 63[1988]-207709

AIR CONDITIONER FOR VEHICLE

Shigenobu Obara

UNITED STATES PATENT AND TRADEMARK OFFICE
WASHINGTON, D.C. JULY 2005
TRANSLATED BY THE MCELROY TRANSLATION COMPANY

JAPANESE PATENT OFFICE
PATENT JOURNAL (BX)
KOKAI PATENT APPLICATION NO. SHO 63[1988]-207709

Int. Cl.⁴:	B 60 H 1/00 F 25 B 27/02
Sequence Nos. for Office Use:	C-7153-3L 7501-3L
Filing No.:	Shop 62[1987]-39649
Filing Date:	February 23, 1987
Publication Date:	August 29, 1988
No. of Claims:	1 (Total of 7 pages)
Examination Request:	Not filed

AIR CONDITIONER FOR VEHICLE

[Jidoshayo kuchosochi]

Inventor:	Shigenobu Obara
Applicant:	Toyota Motor Corp.

[There are no amendments to this patent.]

Claim /61*

An air conditioner for vehicle characterized by the fact that it has automatic control of the air conditioner, which connects a compressor, a condenser, and an evaporator by means of refrigerant piping, provides a heat exchanger for the heat pump, which heats the aforementioned refrigerant piping with the heat source of the engine to said refrigerant piping, and freely switches between cooling and heating by switching the refrigerant path; wherein a first water valve, which controls the flow rate of the hot water flowing from the engine to the heat

* [*The numbers in the margin indicate pagination in the foreign text.]

exchanger for the heat pump, a heater core, and a second water valve for the heater core were provided to the heater pipeline between the engine and the heat exchanger for the heat pump, operating information on the heat pump side is obtained with a mode sensor, a heat pump sensor, and a water temperature sensor, and the flow rate of the hot water flowing into the heater core and the heat exchanger for the heat pump is regulated by controlling the opening and closing of the aforementioned first and second water valves.

Detailed explanation of the invention

Industrial application field

This invention relates to an air conditioner for vehicle capable of freely switching between heating and cooling operations by switching the refrigerant path.

Prior art

The conventional air conditioner for vehicle with a heat pump structure utilizes a heat source of low temperature, which could not be used in refrigerating devices, for heating by increasing the temperature as shown in Figure 5 and Figure 6. Compressor (1), evaporator (3), and condenser (4) shown in Figure 5 are connected by means of refrigerant piping (2), refrigerant piping (2) is connected to meandering forming part (7) formed into a zigzag shape via receiver tanks (6a) and (6b) from aforementioned refrigerant piping (2), heat exchanger (8) is formed by covering meandering forming part (7) of refrigerant piping (2), and the cooling water of engine (9) is circulated to this heat exchanger (8) according to heater pipeline (10) provided with water valve (130).

And then, during a heating operation, electromagnetic valves (26a), (26b), and (26c) /62
shown in Figure 5 are turned off (closed), electromagnetic valves (26d), (26e), and (26f) are turned on (opened), and the refrigerant fluid within refrigerant piping (2) flows in the solid line arrow direction in Figure 5.

On the other hand, in aforementioned heat exchanger (8), cooling water of high temperature from engine (9) flows into aforementioned heat exchanger (8) via heater core (12) by opening water valve (130) provided to heater pipeline (10) and the refrigerant fluid passing through meandering forming part (7) of refrigerant piping (2) within heat exchanger (8) absorbs heat from the high temperature cooling water in engine (9).

As described above, the refrigerant fluid that absorbed heat is further compressed into a high temperature by compressor (1), this high temperature refrigerant fluid is fed and condensed in evaporator (3) by passing through electromagnetic valve (26d), which is opened, and discharges heat.

Figure 6 shows a schematic configuration of an air conditioner arranged with air blower (23), evaporator (3), and heater core (12) in air conditioning case (22) and feeds the wind from air blower (23) to air outlets (25a)-(25f) leading into the chamber according to opening and closing of dampers (20), (24a), (24b), and (24c). The aforementioned high temperature refrigerant fluid fed to this evaporator (3) condenses and discharges heat hence even evaporator (3) becomes a high temperature in addition to heater core (12), the wind from air blower (23) is heated, and takes on a heating operation.

Also, during a cooling operation, electromagnetic valves (26a), (26b), and (26c) shown in Figure 5 are turned on (opened), electromagnetic valves (26d), (26e), and (26f) are turned off (closed), the refrigerant fluid in refrigerant piping (2) passes through condenser (4), and flows like a regular cooling cycle in the broken line arrow direction in Figure 5.

At this time, water valve (130) of heater pipeline (10) passing through aforementioned heat exchanger (8) is closed and prevents the high temperature cooling water in engine (9) from flowing into heat exchanger (8)

As described above, the air conditioner for vehicle having a heat pump structure shown in Figure 5 utilizes a heat source of low temperature, which could not be used in refrigerating devices employed in coolers, by changing the refrigerant fluid circuit in the cooler cycle and converting into a high temperature.

Problems to be solved by the invention

However, in the case of Figure 5, immediate effectiveness of heat exchanger (8) for the heat pump may decrease according to heat absorption in heater core (12) when the engine water temperature is low if heater core (12) and heat exchanger (8) for the heat pump are used concurrently.

Also, if heater core (12) is used in the vicinity of maximum cool, the quantity of water introduced into heat exchanger (8) for the heat pump minimizes and may cause failure of the compressor from negative pressure operation or output of cold air.

Therefore, in order to solve these problems, this device improves the immediate effectiveness of the heat exchanger for the heat pump by contriving a function for preventing heat absorption by the heater core when the engine water temperature is low.

Means of solving the problems

Therefore, this invention solves the aforementioned problems by providing a first water valve, which controls the flow rate of the hot water flowing into the heat exchanger for the heat pump and a second water valve for the heater core in the heater pipeline between the engine and the heat exchange for the heat pump and controlling the opening and closing of these valves.

To explain more specifically by appending the reference numbers in Figure 1 and Figure 2, it is an air conditioner for vehicle having automatic control of the air conditioner, which connects compressor (1), condenser (4), and evaporator (3) by means of refrigerant piping (2), provides heat exchanger (8) for the heat pump, which heats the aforementioned refrigerant piping (2) with the heat source of engine (9), to said refrigerant piping (2), and freely switches between cooling and heating by switching the refrigerant path; wherein first water valve (11), which controls the flow rate of hot water flowing from engine (9) to heat exchanger (8) for the heat pump, heater core (12), and second water valve (13) for heater core (12) are provided to the heater pipeline (10) between engine (9) and heat exchanger (8) for the heat pump, operating information on the heat pump side is obtained with mode sensor (14), heat pump sensor (15), and water temperature sensor (16), and the flow rate of the hot water flowing into heater core (12) and heat exchanger (8) for the heat pump are adjusted by controlling the opening and closing of the aforementioned /63
first and second water valves (11) and (13).

Operation of the invention

According to the aforementioned means, operating information on the heat pump side is obtained from mode sensor (14), heat pump sensor (15), and water temperature sensor (16) during a control of an automatic air conditioner. The flow rate of the hot water flowing into heat exchanger (8) of the heat pump is controlled according to opening and closing of first water valve (11), the flow rate of the hot water in heater core (12) is controlled according to opening and closing of second water valve (13), and improves the immediate effectiveness of heat exchanger (8) for the heat pump when the water temperature in the engine is low.

Embodiment of the invention

Below, a working example of this invention will be explained based on the appended figures.

The drawings in Figure 1-Figure 4 show a working example of this invention. In the air conditioner for vehicle with a heat pump structure and automatic control of the air conditioner, compressor (1) shown in Figure 2, evaporator (3) connected to this compressor (1) via refrigerant piping (2), and compressor (4) are connected by means of the aforementioned refrigerant piping (2) and to the aforementioned evaporator (3), first expansion valve (5a) is connected to the pipeline of refrigerant piping (2) thereof.

Also, refrigerant piping (2) is connected to meandering forming part (7) formed into a zigzag shape from aforementioned refrigerant piping (2) via receiver tanks (6a) and (6b) and second expansion valve (5b), heat exchanger (8) for the heat pump is formed by covering meandering forming part (7) of this refrigerant piping (2), first water valve (11) is provided to

heater pipeline (10a) between engine (9) and aforementioned heat exchanger (8), the hot water (cooling water) of engine (9) is circulated to aforementioned heat exchanger (8) by opening aforementioned water valve (11), which controls the flow rate of hot water flowing into heat exchanger (8) from engine (9), and the refrigerant in meandering forming part (7) of aforementioned refrigerant piping (2) is heated. Also, bypass line (10b) is provided to aforementioned heater pipeline (10a), heater core (12) is provided to this bypass line (10b), and second water valve (13) for heater core (12) is provided to bypass line (10b) between this heater core (12) and first water valve (11) of aforementioned heater pipeline (10a).

On the other hand, as apparent from Figure 1, mode sensor (14), which discriminates the mode, heat pump sensor (15), which discriminates the on-off of the heat pump switch, and water temperature sensor (16), which discriminates the temperature of the hot water in heat exchanger (8) of the heat pump are provided as sensors for obtaining the operating information on the heat pump side during automatic control of the air conditioner and the operating information on the heat pump side obtained from these sensors (14), (15), and (16) are provided to computer (17) as input signals and drives first and second servomotors (18) and (19).

Aforementioned first servomotor (18) controls the opening and closing of aforementioned first water valve (11) (controls the flow rate of the hot water flowing into heat exchanger (8)) and aforementioned second servomotor (19) controls the opening and closing of aforementioned second water valve (13) (controls the flow rate of hot water flowing into heater core (12)) as well as selectively opening and closing air mix damper (20) to open (hot) or close (cool).

Incidentally, changeover relay (21) is provided between aforementioned computer (17) and second servomotor (19) to changeover the rotation of second servomotor (19).

Also, air blower (23), evaporator (3), and heater core (12) are arranged in air conditioning case (22) shown in Figure 1 and a schematic configuration of an air conditioner which supplies wind from air blower (23) to air outlets (25a)-(25f) leading into the chamber according to opening and closing of dampers (20), (24a), (24b), (24c), (24d), (24e), and (24f) is shown.

And then, during a heating operation, electromagnetic valves (26a), (26b), and (26c) shown in Figure 2 are turned off (closed), electromagnetic valves (26d), (26e), and (26f) are turned on (opened), the refrigerant fluid in refrigerant piping (2) flows in the solid line arrow direction in Figure 2, and even the air conditioner operates in a heating state, hence the refrigerant fluid that heat absorbed in heat exchanger (8) for the heat pump is further compressed into a high temperature according to compressor (1), this high temperature refrigerant fluid is fed to evaporator (3) via electromagnetic valve (26d), condensed, and discharges heat. /64

As described above, the aforementioned high temperature refrigerant fluid fed to evaporator (3) condenses and discharges heat, hence even evaporator (3) takes on a high

temperature in addition to heater core (12), the wind from air blower (23) in Figure 1 is heated, and takes on a dehumidifying heating operation.

Also, during a cooling operation, electromagnetic valves (26a), (26b), and (26c) shown in Figure 2 are turned on (opened) and electromagnetic valves (26d), (26e), and (26f) are turned off (closed) to switch the refrigerant path and the refrigerant fluid in refrigerant piping (2) flows as a regular cooling cycle in the broken line arrow direction in Figure 2 by passing through condenser (4).

At this time, first water valve (11) of heater pipeline (10a) passing through aforementioned heat exchanger (8) is closed and the high temperature cooling water of engine (9) does not flow into heat exchanger (8) hence the refrigerant fluid flows into evaporator (3) and takes on a cooling operation.

Incidentally, reference numbers (27a), (27b), and (27c) in Figure 2 are the check valves.

In Figure 4, temperature sensor (29) and compressor switch (30) were provided in the connection circuit between compressor (1) and power source (28). Control of compressor (1) during a heating operation is outlet temperature control of evaporator (3) and discharge pressure control of compressor (1), temperature sensor (29) of evaporator (3) turns off compressor (1) when the outlet temperature thereof is 80°C or greater, turns on at 70°C or less, compressor switch (30) turns off compressor (1) when the discharge pressure of compressor (1) is 20 kg/cm^2 or greater, and turns it on when it is 13, 5 kg/cm^2 [TN: as is in the foreign text] or less.

Incidentally, reference number (31) in Figure 1 is the inside sensor, reference number (32) the outside air sensor, reference number (33) the solar radiation sensor, reference number (34) the temperature setting lever, and reference number (35) the blower changeover switch.

Figure 3 shows a working example of this invention as a flow chart, signals of mode sensor (14) and heat pump sensor (15) and the signals of water temperature sensor (16) of engine (9) are received when carrying out a warm up control during automatic control of the air conditioner and air mix damper (20) of heater core (12) and second water valve (13) are opened or closed by operating changeover relay (21).

Also, first water valve (11) of heat exchanger (8) for the heat pump is also opened or closed concurrently.

Effects of the invention

As was described above, this invention provides a first water valve, which controls the flow rate of the hot water flowing into the heat exchanger for the heat pump and a second water valve for the heater core to the heater pipeline between the engine and the heat exchange for the heat pump and the water path in the heat exchange for the heat pump and the heater core is changed by controlling the opening and closing of these valves according to a mode sensor, a

heat pump sensor, and a water temperature sensor, hence heat absorption by the heater core is prevented when the water temperature in the engine is low, the immediate effectiveness in the heat exchange for the heat pump is improved, and failure of the compressor during negative pressure operation or discharge of cold air can be prevented.

Also, it is necessary to use the heater core at maximum cool when the heat pump is in use but this invention can solve the inconveniences by combining with an automatic air conditioner.

Brief description of the figures

The drawings from Figure 1 to Figure 4 show a working example of this invention,

Figure 1 is a block diagram of the control system of the air conditioner,

Figure 2 is a schematic diagram of the refrigerating cycle,

Figure 3 is a flow chart showing the essence of this invention,

Figure 4 is a connection diagram that connects the compressor to the power source via sensors and switches,

Figure 5 and Figure 6 show a conventional example, Figure 5 is a block diagram corresponding to Figure 2, and

Figure 6 is a schematic diagram of an air conditioner.

(1)...compressor, (2)...refrigerant piping, (3)...evaporator, (4)...condenser, (8)...heat exchanger, (9)...engine, (10a)...heater pipeline, (11)...first water valve, (12)...heater core, (13)...second water valve, (14)...mode sensor, (15)...heat pump sensor, (16)...water temperature sensor.

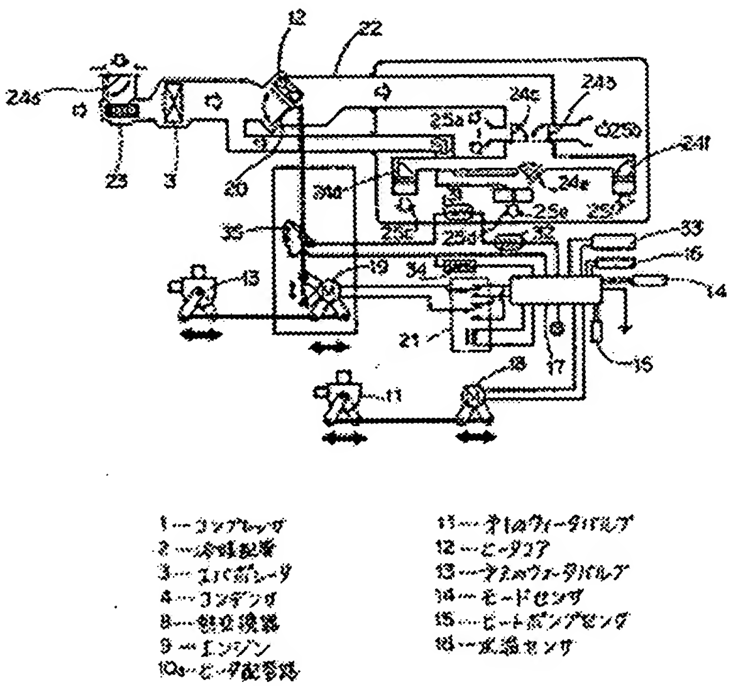

Figure 1

Key:

1	Compressor
2	Refrigerant piping
3	Evaporator
4	Condenser
8	Heat exchanger
9	Engine
10a	Heater pipeline
11	First water valve
12	Heater core
13	Second water valve
14	Mode sensor
15	Heat pump sensor
16	Water temperature sensor

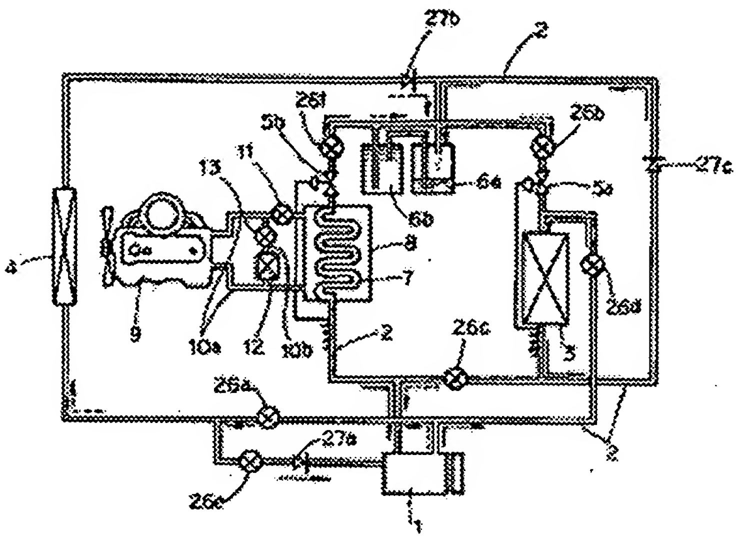

Figure 2

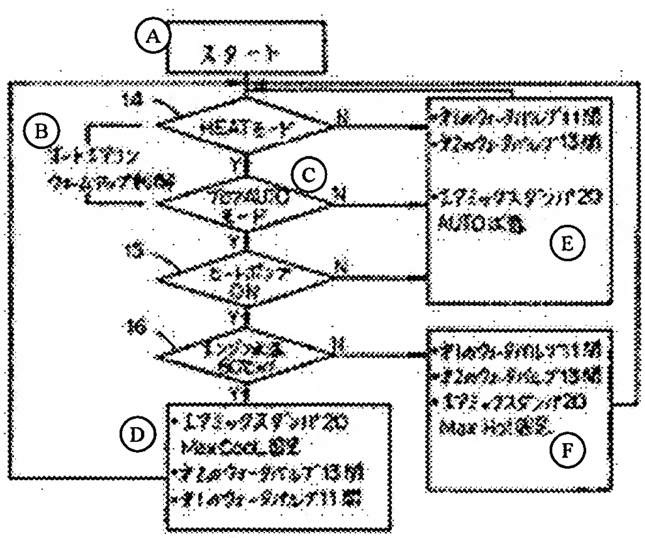

Figure 3

Key:

A	Start
B	Auto air conditioner warm up control
C	Blower Auto mode
D	•Air mix damper (20) fixed at Max Cool •Second water valve (13) closed •First water valve (11) opened
E	•First water valve (11) closed •Second water valve (13) opened •Air mix damper (20) in the AUTO state
F	•First water valve (11) [illegible – closed?] •Second water valve (13) opened •Air mix damper (2) fixed at Max Hot
14	Heat mode

15 Heat pump on
16 Engine water temperature
60°CWF

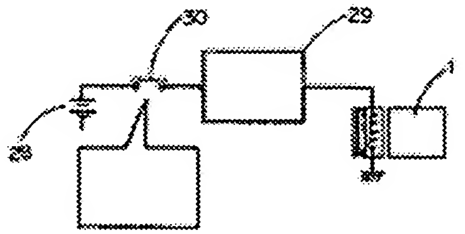

Figure 4

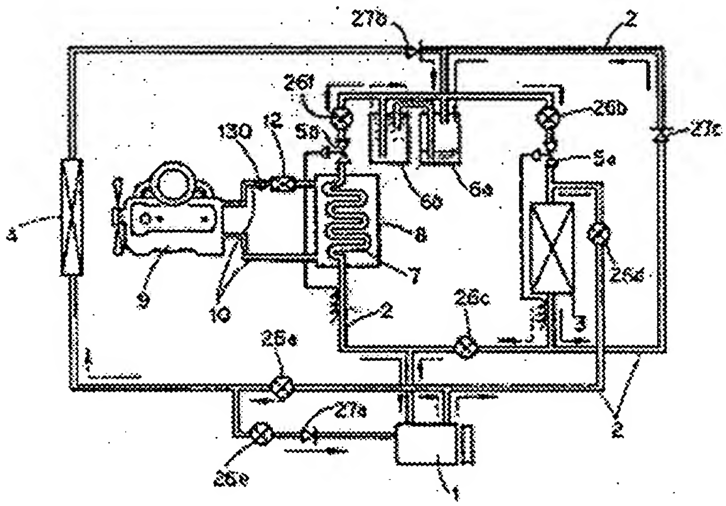

Figure 5

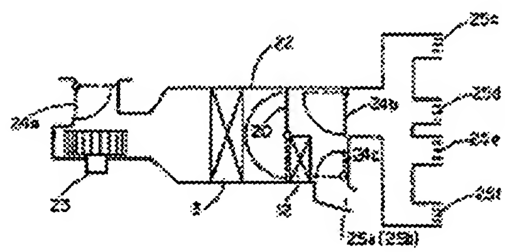

Figure 6

copy of JP 11-34640
with machine translation
and US equivalent
(6,047,770)

(19)日本国特許庁（ＪＰ）　(12) 公開特許公報(A)　(11)特許出願公開番号

特開平11－34640

(43)公開日　平成11年(1999) 2月9日

(51)Int.Cl.⁶　識別記号　ＦＩ

(51)Int.Cl.⁶	識別記号	ＦＩ	
Ｂ６０Ｈ　1/03		Ｂ６０Ｈ　1/03	Ｚ
1/22		1/22	

審査請求　未請求　請求項の数7　ＯＬ　(全 10 頁)

(21)出願番号　特願平9－198831

(22)出願日　平成9年(1997) 7月24日

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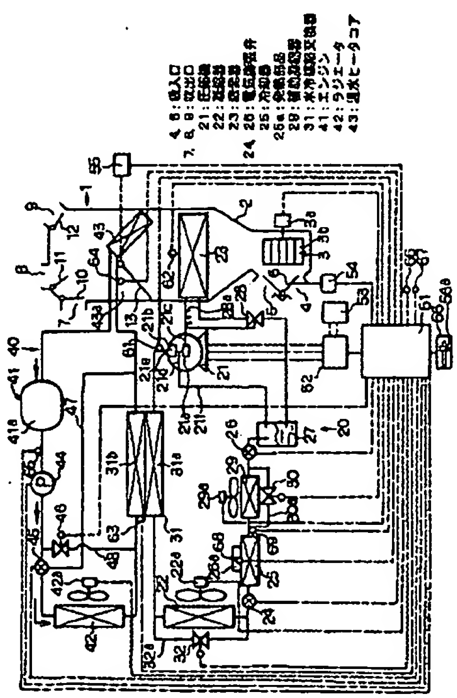
最終頁に続く

(54)【発明の名称】　車両用空調装置

(57)【要約】

【課題】　冬期には、車載発熱部品の冷却熱を有効利用して暖房を可能とし、一方、夏期には車載発熱部品の冷却を十分可能とする。

【解決手段】　冷凍サイクル２０の圧縮機２１吐出側に、圧縮機吐出ガス冷媒により冷却水を加熱する水冷媒熱交換器３１を設け、サイクル中間圧力の冷媒により車載発熱部品２５ａを冷却する冷却器２５を設ける。空調ダクト２には蒸発器２３および温水ヒータコア４３を設置する。冷却水回路４０には水冷媒熱交換器３１と温水ヒータコア４３の他に、エンジン４１とラジエータ４２と電動ポンプ４４とを備える。車載発熱部品２５ａの冷却熱を圧縮機２１で汲み上げた後、冷却水に放熱することができ、高外気温時でも、発熱部品を十分冷却できる。



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【特許請求の範囲】

【請求項1】 一端側に空気の吸入口（４、５）を有し、他端側に車室内への吹出口（７、８、９）を有する空調空気通路（２）と、

この空調空気通路（２）に設置され、この空調空気通路（２）を通して空気を前記吸入口（４、５）側から前記吹出口（７、８、９）側へ送風する送風機（３）と、

冷凍サイクル低圧側の冷媒を吸入、圧縮して吐出する圧縮機（２１）と、

この圧縮機（２１）から吐出されたガス冷媒と冷却水との間で熱交換を行って、この冷却水を加熱する水冷媒熱交換器（３１）と、

この水冷媒熱交換器（３１）を通過した冷媒を外気との間で熱交換して、凝縮する凝縮器（２２）と、

前記水冷媒熱交換器（３１）および前記凝縮器（２２）の少なくとも一方で凝縮した高圧の液冷媒を中間圧力まで減圧する高圧側の減圧手段（２４）と、

前記中間圧力の冷媒を車両搭載の発熱部品（２５ａ）から吸熱して蒸発させ、この発熱部品（２５ａ）を冷却するように構成された冷却器（２５）と、

この冷却器（２５）通過後の中間圧力の冷媒を低圧に減圧する低圧側の減圧手段（２８）と、

前記空調空気通路（２）に設置され、前記低圧の冷媒を前記空気から吸熱して蒸発させ、前記空気を冷却する蒸発器（２３）と、

前記空調空気通路（２）において、前記蒸発器（２３）の空気下流側に設置され、前記空気を前記冷却水と熱交換して加熱する温水ヒータコア（４３）とを備え、

さらに、この温水ヒータコア（４３）に前記冷却水を循環する冷却水回路（４０）に、車両の駆動力を直接または間接的に得るためのエンジン（４１）と、前記冷却水を外気との間で熱交換して冷却するラジエータ（４２）と、前記冷却水を前記冷却水回路（４０）に循環させるポンプ（４４）と、前記水冷媒熱交換器（３１）とを備えることを特徴とする車両用空調装置。

【請求項2】 一端側に空気の吸入口（４、５）を有し、他端側に車室内への吹出口（７、８、９）を有する空調空気通路（２）と、

この空調空気通路（２）に設置され、この空調空気通路（２）を通して空気を前記吸入口（４、５）側から前記吹出口（７、８、９）側へ送風する送風機（３）と、

冷凍サイクル低圧側の冷媒を吸入ポート（２１ｃ）から吸入し中間圧力まで圧縮する低段側圧縮部（２１ａ）と、前記中間圧力まで圧縮されたガス冷媒とガスインジェクションポート（２１ｄ）から流入するガス冷媒との混合ガスを吐出圧力まで圧縮して吐出ポート（２１ｅ）から吐出する高段側圧縮部（２１ｂ）とを有する圧縮機（21）と、

この圧縮機（２１）から吐出されたガス冷媒と冷却水との間で熱交換を行って、冷却水を加熱する水冷媒熱交換器（３１）と、

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この水冷媒熱交換器（３１）を通過した冷媒と外気との間で熱交換を行って、冷媒を凝縮する凝縮器（２２）と、

前記水冷媒熱交換器（３１）および前記凝縮器（２２）の少なくとも一方で凝縮した高圧の液冷媒を中間圧力まで減圧する高圧側の減圧手段（２４）と、

前記中間圧力の冷媒を車両搭載の発熱部品（２５ａ）から吸熱して蒸発させ、この発熱部品（２５ａ）を冷却するように構成された冷却器（２５）と、

この冷却器（２５）の下流側に設けられ、前記中間圧力の冷媒の気液を分離する気液分離器（２７）と、

この気液分離器（２７）で分離された液冷媒を減圧する低圧側の減圧手段（２８）と、

前記空調空気通路（２）に設置され、前記低圧の冷媒を前記空気から吸熱して蒸発させ、前記空気を冷却する蒸発器（２３）と、

前記気液分離器（２７）で分離されたガス冷媒を前記ガスインジェクションポート（２１ｄ）に導くガスインジェクション通路（２１ｆ）と、

前記空調空気通路（２）において、前記蒸発器（２３）の空気下流側に設置され、前記空気を前記冷却水と熱交換して加熱する温水ヒータコア（４３）とを備え、

さらに、この温水ヒータコア（４３）に前記冷却水を循環する冷却水回路（４０）に、車両の駆動力を直接または間接的に得るためのエンジン（４１）と、前記冷却水を外気と熱交換して冷却するラジエータ（４２）と、前記冷却水を前記冷却水回路（４０）に循環させるポンプ（４４）と、前記水冷媒熱交換器（３１）とを備えることを特徴とする車両用空調装置。

【請求項3】 前記冷却器（２５）の下流側に配置され、前記中間圧力の冷媒を外気と熱交換して凝縮させる補助凝縮器（２９）と、

この補助凝縮器（２９）をバイパスするバイパス通路（３０ａ）と、

このバイパス通路（３０ａ）を開閉する開閉弁（３０）と、

前記中間圧力の冷媒の温度に相当する物理量を検出する検出手段（６９）と、

外気温度を検出する検出手段（６６）とを備え、

前記中間圧力の冷媒の温度が外気温度より低い場合は、前記開閉弁（３０）を開いて、前記中間圧力の冷媒を前記補助凝縮器（２９）をバイパスして流すことを特徴とする請求項１または２に記載の車両用空調装置。

【請求項4】 少なくとも冷房運転時には、前記ラジエータ（４２）で冷却された冷却水が前記水冷媒熱交換器（３１）に流入するようにしたことを特徴とする請求項１ないし３のいずれか１つに記載の車両用空調装置。

【請求項5】 少なくとも暖房運転時には前記水冷媒熱交換器（３１）で加熱された冷却水が前記温水ヒータコ

ア(43)に流入するようにしたことを特徴とする請求項1ないし4のいずれか1つに記載の車両用空調装置。

【請求項6】　前記ラジエータ(42)をバイパスして冷却水を流すバイパス通路(48)と、このバイパス通路(48)を開閉する開閉弁(46)と、前記冷却水回路(40)の冷却水温度を検出する検出手段(65)とを備え、

暖房運転時に前記冷却水温度が所定値より低い場合は、前記開閉弁(46)を開いて、前記ラジエータ(42)をバイパスして前記冷却水を流すことを特徴とする請求項1ないし5のいずれか1つに記載の車両用空調装置。

【請求項7】　前記ラジエータ(42)を通過する冷却水流量と前記ラジエータ(42)をバイパスする冷却水流量の比率を冷却水温度に応じて制御する流量制御弁(45)を備えることを特徴とする請求項1ないし5のいずれか1つに記載の車両用空調装置。

【発明の詳細な説明】

【0001】

【発明の属する技術分野】本発明は電気自動車(EV)、ハイブリッド車(HV)用として好適な車両用空調装置に関するもので、特に、車載発熱部品の冷却とその冷却熱(廃熱)の有効利用に関する。

【0002】

【従来の技術】従来よりEV、HV用の空調装置として、冬期に車載発熱部品の冷却熱を有効利用して暖房を行うシステムが提案されている。例えば、特開平8-258548号公報には車載発熱部品の冷却熱をヒートポンプの吸熱源として用いて、暖房に利用する空調装置が示されている。また、特開平8-197937号公報には車載発熱部品の冷却熱で暖められた冷却水をヒートポンプにてさらに加熱して暖房を行う空調装置が示されている。

【0003】

【発明が解決しようとする課題】しかし、これらの従来装置では、いずれも、夏期には車載発熱部品の冷却を冷却水で行い、この冷却水をラジエータに循環し、ラジエータにて冷却水の熱を外気に放熱しているため、外気温度が40℃を越えるような夏期高温時には冷却水温度が65℃程度に上昇するため、車載発熱部品の冷却を十分に行うことができないという問題がある。

【0004】本発明は上記点に鑑みて、冬期には、車載発熱部品の冷却熱を有効利用して暖房を可能とし、一方、夏期には車載発熱部品の冷却を十分可能にする車両用空調装置を提供することを目的とする。

【0005】

【課題を解決するための手段】上記目的を達成するために、請求項1に記載の発明では、冷凍サイクルの圧縮機(21)吐出側に、圧縮機吐出ガス冷媒と冷却水との間で熱交換を行って、この冷却水を加熱する水冷媒熱交換器(31)を設けるとともに、サイクル高圧側の液冷媒を高圧側の減圧手段(24)で中間圧力まで減圧し、この中間圧力の冷媒が流入する冷却器(25)を設け、この冷却器(25)において中間圧力の冷媒が車両搭載の発熱部品(25a)から吸熱して蒸発することにより、発熱部品(25a)の冷却を行い、空調空気通路(2)には、サイクル低圧側の冷媒を空気から吸熱して蒸発させ、空気を冷却する蒸発器(23)を設置し、空調空気通路(2)において、蒸発器(23)の空気下流側に、空気を冷却水と熱交換して加熱する温水ヒータコア(43)を設置し、さらに、この温水ヒータコア(43)に冷却水を循環する冷却水回路(40)に、車両の駆動力を直接または間接的に得るためのエンジン(41)と、冷却水を外気との間で熱交換して冷却するラジエータ(42)と、冷却水を冷却水回路(40)に循環させるポンプ(44)と、水冷媒熱交換器(31)とを備えることを特徴としている。

【0006】これによると、車両空調用冷凍サイクルの中間圧力冷媒で車載発熱部品(25a)を冷却するとともに、冷凍サイクルの圧縮機吐出冷媒と冷却水とを熱交換する水冷媒熱交換器(31)を用いることにより、車載発熱部品(25a)の冷却熱を圧縮機(21)で汲み上げた後、冷却水に放熱することができる。従って、夏期の高外気温時であっても、中間圧力冷媒で十分に車載発熱部品(25a)を冷却することができる。

【0007】しかも、夏期冷房時の冷凍サイクルの放熱を、凝縮器(22)だけでなく、水冷媒熱交換器(31)を通してラジエータ(42)でも放熱することができるため、サイクル全体としての放熱能力を向上できる。そのため、高圧の上昇を抑制して圧縮機(21)の消費動力(消費電力)を低減できる。さらに、冬期暖房時には車両搭載の発熱部品(25a)の廃熱を利用して水冷媒熱交換器(31)を通して冷却水を加熱し、この冷却水を用いて効果的に暖房を行うことが可能となる。従って、エンジン廃熱がない場合でも暖房が可能となる。

【0008】また、請求項2記載の発明では、請求項1記載の発明に対して、ガスインジェクション機能を持つ冷凍サイクルを構成しているから、ガスインジェクション機能による圧縮機(21)での圧縮比の低下による圧縮動力の低減、水冷媒熱交換器(31)への冷媒循環量増大による暖房能力増大等の効果を発揮できる。また、請求項3記載の発明では、車載発熱部品(25a)の冷却器(25)の下流側に、中間圧力の冷媒を外気と熱交換して凝縮させる補助凝縮器(29)およびこの補助凝縮器(29)をバイパスするバイパス通路(30a)を配置し、このバイパス通路(30a)に開閉弁(30)を配置し、中間圧力の冷媒の温度に相当する物理量を検出する検出手段(69)と、外気温度を検出する検出手段(66)とを備え、中間圧力の冷媒の温度が外気温度より低い場合は、開閉弁(30)を開いて、中間圧力の

冷媒を前記補助凝縮器（２９）をバイパスして流すことを特徴としている。

【０００９】これによると、補助凝縮器（２９）において、中間圧力の冷媒が外気から吸熱してサイクル効率を悪化させるという不具合を確実に防止でき、一方、中間圧力の冷媒を補助凝縮器（２９）にて冷却、凝縮できる条件のときは、補助凝縮器（２９）による冷媒凝縮作用によりサイクル能力を増大できる。また、請求項４記載の発明では、少なくとも冷房運転時には、ラジエータ（４２）で冷却された冷却水が水冷媒熱交換器（３１）に流入するようにしたことを特徴としている。

【００１０】これによると、ラジエータ（４２）で冷却された低温の冷却水にて水冷媒熱交換器（３１）における吐出ガス冷媒を効率よく冷却できる。また、請求項５記載の発明では、少なくとも暖房運転時には水冷媒熱交換器（３１）で加熱された冷却水が温水ヒータコア（４３）に流入するようにしたことを特徴としている。

【００１１】これによると、水冷媒熱交換器（３１）で加熱された冷却水が冷却水回路（４０）の他の箇所で放熱することなく、直ちに温水ヒータコア（４３）に流入するから、効果的に暖房能力を向上できる。また、請求項６記載の発明では、ラジエータ（４２）をバイパスして冷却水を流すバイパス通路（４８）と、このバイパス通路（４８）を開閉する開閉弁（４６）と、冷却水回路（４０）の冷却水温度を検出する検出手段（６５）とを備え、暖房運転時に冷却水温度が所定値より低い場合は、開閉弁（４６）を開いて、ラジエータ（４２）をバイパスして冷却水を流すことを特徴としている。

【００１２】これによると、冷却水温度の低いときに、ラジエータ（４２）における冷却水の放熱を防止して、暖房能力を確保できる。また、請求項７記載の発明では、ラジエータ（４２）を通過する冷却水流量とラジエータ（４２）をバイパスする冷却水流量の比率を冷却水温度に応じて制御する流量制御弁（４５）を備えることを特徴としている。

【００１３】これによると、流量制御弁（４５）の作用にてラジエータ（４２）への冷却水通過流量の比率を制御することにより、冷房、暖房の運転条件の変化に対応した、適切な冷却水温度を設定できる。なお、上記各手段および特許請求の範囲に記載の各手段に付した括弧内の符号は、後述する実施形態記載の具体的手段との対応関係を示すものである。

【００１４】

【発明の実施の形態】以下、本発明を図に示す実施形態について説明する。図１は本発明の一実施形態による車両用空調装置の全体システム構成を示す。図１は、車両走行駆動源としてエンジン（内燃機関）４１と、モータ（図示せず）の両方を具備するハイブリッド車（ＨＶ）に本発明を適用した場合を示している。車両用空調ユニット１は、車両の車室内前部の計器盤下部に設置されるものであって、その空調ダクト２は、車室内に空調空気を導く空調用通路を構成している。

【００１５】この空調ダクト２の一端側には内外気を吸入する吸入口４、５が設けられている。内気吸入口４と外気吸入口５は内外気切替ドア６により切替開閉される。この内外気切替ドア６はサーボモータ５４により開閉制御される。上記吸入口４、５に隣接して、空調ダクト２内に空気を送風する送風機３が設置されており、この送風機３はモータ３ａとこのモータ３ａにより駆動される遠心ファン３ｂとにより構成されている。一方、空調ダクト２の他端側には車室内へ通ずる複数の吹出口７、８、９が形成されている。これらの吹出口７、８、９は吹出モード切替ドア１０、１１、１２によりそれぞれ切替開閉され、フェイス、バイレベル、フット、デフロスタ等の吹出モードが設定される。

【００１６】また、送風機３より空気下流側における空調ダクト２内には冷凍サイクル２０の蒸発器２３が設けられている。この蒸発器２３は、冷凍サイクルの低圧冷媒が空気から吸熱して、空気を冷却するものである。さらに、蒸発器２３より空気下流側には冷却水回路４０の温水ヒータコア４３が設けられており、この温水ヒータコア４３は冷却水（温水）を熱源として空気を加熱するものである。

【００１７】この温水ヒータコア４３の側方にはバイパス路４３ａが形成され、また、温水ヒータコア４３の空気入口部にはエアミックスドア１３が回動自在に配設されている。このエアミックスドア１３は、温水ヒータコア４３を通過する空気（温風）の流れと、バイパス路４３ａを通過して温水ヒータコア４３をバイパスする空気（冷風）の流れとを調整する。このエアミックスドア１３はサーボモータ５５により開度（回動量）が制御される。

【００１８】次に、冷凍サイクル２０の構成について説明する。冷凍サイクルは前記蒸発器２３の他に、以下の機器により構成されている。すなわち、圧縮機２１はモータ（図示せず）により駆動される電動式であり、圧縮機２１の吐出側には水冷媒熱交換器３１が設けられている。この水冷媒熱交換器３１には、圧縮機２１の吐出ガス冷媒が流れる冷媒流路３１ａと後述の冷却水回路４０の冷却水が流れる冷却水通路３１ｂが設けられ、冷媒流路３１ａの圧縮機吐出ガス冷媒と、冷却水通路３１ｂの冷却水との間で熱交換を行うようになっている。

【００１９】この水冷媒熱交換器３１の下流側には凝縮器２２が設けられ、ここで、ガス冷媒は外気と熱交換を行って冷却され、凝縮する。この凝縮器２２には並列にバイパス通路３２ａが設けられ、このバイパス通路３２ａを電磁弁３２により開閉する。この凝縮器２２とバイパス通路３２ａの並列回路の下流側には第１電気膨張弁（第１減圧手段、高圧側減圧手段）２４が設けられ、この第１電気膨張弁２４により高圧冷媒を第１中間圧力ま

で減圧する。第1電気膨張弁24は第1中間圧力が目標圧力となるように弁開度（絞り量）が電気的に制御される。

【0020】そして、第1電気膨張弁24の下流側に冷却器25が設けられ、この冷却器25は第1中間圧力冷媒にて車両搭載の発熱部品25aを冷却する。ここで、車両搭載の発熱部品25aとしては、例えばHV、EV車両の走行用モータおよびその回転数制御用インバータの半導体スイッチ素子（パワートランジスタ）、車載バッテリ等である。

【0021】冷却器25の下流側に補助凝縮器29が設けられ、この補助凝縮器29は、第1中間圧冷媒と外気との間で熱交換を行うことにより、第1中間圧冷媒を凝縮する。この補助凝縮器29にもバイパス通路30aが並列に設けられ、このバイパス通路30aを電磁弁30にて開閉する。この補助凝縮器29と電磁弁30の並列回路の下流側に第2電気膨張弁（第2減圧手段）26が設けられ、この第2電気膨張弁26は第1中間圧力冷媒をさらに第2中間圧力まで減圧する。この第2電気膨張弁26は第2中間圧力が目標圧力となるように弁開度（絞り量）が電気的に制御される。

【0022】第2電気膨張弁26の下流側に気液分離器27が設けられ、この気液分離器27は第2中間圧力の冷媒の気液分離を行うと共に液冷媒を溜める機能を果たす。この気液分離器27で気液分離された液冷媒は、温度式膨張弁（第3減圧手段、低圧側減圧手段）28により低圧圧力まで減圧される。この温度式膨張弁28は蒸発器23の出口冷媒（圧縮機吸入冷媒）の温度を感知する感温筒28aを有し，蒸発器出口冷媒の過熱度が所定値に維持されるように弁開度（冷媒流量）を調整する。

【0023】圧縮機21は、冷凍サイクル低圧側の冷媒を吸入ポート21cから吸入し中間圧力まで圧縮する低段側圧縮部21a（図2のモリエル線図参照）と、中間圧力まで圧縮されたガス冷媒とガスインジェクションポート21dから流入するガス冷媒の混合ガスを吐出圧力まで圧縮し吐出ポート21eから吐出する高段側圧縮部21bとからなるガスインジェクション型圧縮機で、例えば、スクロール圧縮機からなる。

【0024】また、冷媒圧縮機21のモータにはインバータ52により交流電圧が印加され、このインバータ52により交流電圧の周波数を調整することによってモータ回転速度を連続的に変化させるようになっている。このインバータ52には車載バッテリ53からの直流電圧が印加される。一方、冷却水回路40には、直接および間接的に車両の駆動源となるエンジン（内燃機関）41の冷却部41aが設けられている。ここで、エンジン41はその動力を直接、機械的に取り出して車両の走行駆動源として利用する場合の他に、エンジン41を発電機駆動専用として用いる場合もある。この場合は発電機の発電出力にて走行用モータの駆動および車載バッテリ53への充電を行って、車両の走行を行う。

【0025】冷却水回路40には、エンジン41の停止時も車載バッテリ53を電源として駆動可能な電動ウォータポンプ44、冷却水を外気と熱交換して冷却するラジエータ42、前記した水冷媒熱交換器31の冷却水流路31b、および前記温水ヒータコア43がエンジン41と直列に配置されて、冷却水の循環する閉回路を構成する。電動ウォータポンプ44は、エンジン41からラジエータ42に向かう方向（矢印方向）に冷却水を圧送する。

【0026】また、ラジエータ42と水冷媒熱交換器31の冷却水流路31bをバイパスする第1バイパス通路47を設けるとともに、ラジエータ42と流量制御弁45をバイパスする第2バイパス通路48が設けられている。流量制御弁45はラジエータ42の入口部と第1バイパス通路47の入口部との分岐点に設けられている。この流量制御弁45は三方弁タイプのもので、ラジエータ42で放熱をする冷却水量とラジエータ42をバイパスする冷却水量の比率を連続的（リニア）に可変制御可能なものである。このため、流量制御弁45は弁開度を電気的に調整可能に構成されている。一方、第2バイパス通路48には通路を開閉する電磁弁46が設けられている。

【0027】ところで、上記の流量制御弁45、電磁弁46等の電気機器を通電制御するために空調制御装置51が備えられており、この空調用制御装置51はマイクロコンピュータとその周辺回路にて構成される電子制御装置である。空調用制御装置51は、上記弁類の他に、インバータ52、内外気ドア用サーボモータ54、エアミックスドア用サーボモータ55、電動ウォータポンプ44、送風機3、外気送風ファン22a、29a、42a等の通電制御も行う。

【0028】空調用制御装置51には次の各種センサ等から入力信号が加えられる。すなわち、圧縮機吐出ガス冷媒の温度を検出する吐出温度センサ61、蒸発器23の吹出直後の空気温度を検出する蒸発器後温度センサ62、水冷媒熱交換器31の冷却水入口温度を検出する水冷媒入口温度センサ63、温水ヒータコア43の入口水温を検出するヒータコア入口水温センサ64、エンジン冷却部41aの出口水温を検出するエンジン水温センサ65、外気温度を検出する外気温センサ66、車室内の空気温度を検出する内気温度センサ67、発熱部品25aの冷却温度を検出する冷却温度センサ68、発熱部品冷却器25の冷媒温度を検出する中間圧冷媒温度センサ69、および空調コントロールパネル56の温度制御レバー56a等の各種レバー、スイッチからの空調操作信号が空調用制御装置51に入力される。

【0029】次に、上記構成において作動を説明する。図3は空調用制御装置51による制御ルーチンを示すもので、空調コントロールパネル56の空調作動スイッチ

（図示せず）の投入により制御ルーチンが起動し、ステップ１００において、空調装置が冷房運転にあるか暖房運転にあるかを判定する。この判定は、例えば、空調コントロールパネル５６の温度制御レバー５６ａの操作位置が低温側（冷房側）にあるか、高温側（暖房側）にあるかによって行う。

【００３０】ステップ１００において冷房運転が判定されると、ステップ１０１に進み、冷凍サイクル２０の電磁弁３２および冷却水回路４０の電磁弁４６を閉弁する。冷房運転時において、エンジン４１が通常の負荷条件で作動している時には、ラジエータ４２の放熱性能により水冷媒熱交換器３１に流入する冷却水温度が概略６０゜Ｃ付近であり、一方、圧縮機２１から吐出されるガス冷媒温度は通常８０゜Ｃ付近である。

【００３１】従って、水冷媒熱交換器３１において、冷却水温度が冷媒温度より低くなっているので、圧縮機２１から吐出された高温高圧のガス冷媒は水冷媒熱交換器３１で冷却水に放熱を行い、冷却される。その結果、圧縮機吐出ガス冷媒の一部は水冷媒熱交換器３１の冷媒通路３１ａにて凝縮する。ここで、冷房時には上述のごとくステップ１０１にて、冷凍サイクルの電磁弁３２が閉じているため、水冷媒熱交換器３１の冷媒通路３１ａを出た冷媒は凝縮器２２で外気と熱交換を行い、冷却され、凝縮する。

【００３２】この高温、高圧の液冷媒は、次に第１電気膨張弁２４で第１中間圧力まで減圧され、気液二相状態となる。そして、この気液二相冷媒は冷却器２５に流入して車両搭載の発熱部品２５ａを冷却する。すなわち、冷却器２５において、気液二相冷媒中の液冷媒は車両搭載の発熱部品２５ａから吸熱して蒸発するとともに発熱部品２５ａを冷却する。

【００３３】ここで、冷却温度センサ６８により検出される発熱部品２５ａの冷却温度が所定値（例えば外気温度＋5℃）となるよう、発熱部品２５ａの発熱量に応じて第１中間圧力冷媒温度が第１電気膨張弁２４により制御される。つまり、発熱部品２５ａの発熱量が大きいときは冷却量を増やす必要があるため、第１電気膨張弁２４の開度を小さくして（絞り量を大きくして）第１中間圧力（冷媒温度）を低くする。

【００３４】一方、発熱部品２５ａの発熱量が小さいときは冷却量は少なくてよいため、第１電気膨張弁２４の開度を大きくして（絞り量を小さくして）第１中間圧力（冷媒温度）を高くする。これにより、車両搭載の発熱部品２５ａの発熱量、外気温度が変化しても、常に必要な量だけ発熱部品２５ａを適切に冷却することができる。

【００３５】一方、中間圧冷媒温度センサ６９で検出された冷媒温度が外気温度より高い場合には、図3のステップ１０２の判定がＹＥＳとなり、ステップ１０３に進み電磁弁３０を閉じる。従って、発熱部品冷却器２５から出た冷媒は、補助凝縮器２９に流入し、外気と熱交換を行い再び冷却され、凝縮する。これに反し、第１中間圧力の冷媒温度が外気温度より低い場合には、ステップ１０２からステップ１０４に進み、電磁弁３０を開く。そのため、第１中間圧力冷媒はバイパス通路３０ａを通過して補助凝縮器２９をバイパスし、第２電気膨張弁２６に流入する。このように、第１中間圧力の冷媒温度が外気温度より低い場合には第１中間圧力冷媒がバイパス通路３０を通過して補助凝縮器２９をバイパスすることにより、第１中間圧力冷媒が外気から吸熱するという不具合を未然に防止できる。

【００３６】そして、第２電気膨張弁２６で第１中間圧力冷媒は第２中間圧力まで減圧された後、気液分離器２７に流入する。気液分離器２７で二相冷媒はガスと液に分離されガス冷媒はガスインジェクション配管２１ｆを通り、ガスインジェクションポート２１ｄより圧縮機２１の圧縮過程の途中に吸入される。一方、気液分離器２７内の液冷媒は温度式膨張弁２８で低圧圧力まで減圧され蒸発器２３に流入する。なお、温度式膨張弁２８では蒸発器２３の出口で冷媒が完全に蒸発するよう冷媒流量を制御している。

【００３７】空調ユニット１において、送風機3により送風される空気は蒸発器２３で冷媒と熱交換を行って冷却され、車室内の冷房を行う。ここで、冷媒は蒸発器２３で蒸発してガス化し、そのガス冷媒は、吸入ポート２１ｃより圧縮機２１に吸入されて、圧縮される。図2のモリエル線図は、上記した冷凍サイクルの各機器における冷媒の状態を示している。

【００３８】次に、冷房時における冷却水回路４０の作動を図4に基づいて説明すると、図４の横軸はエンジン水温センサ６５により検出されるエンジン冷却部４１ａの出口水温をとり、縦軸は流量制御弁４５により制御されるラジエータ通過流量の比率（ラジエータ通過流量／流量制御弁４５への全流量）である。この図４の制御特性（マップ）は、予め設定され、空調用制御装置５１のマイクロコンピュータのＲＯＭに記憶されているものであって、冷房時にはラジエータバイパス通路４８の電磁弁４６が閉じているので、流量制御弁４５によりラジエータ通過流量を制御できる。

【００３９】そこで、図3のステップ１０５において、図４の制御特性に従ってエンジン冷却部４１ａの出口水温に対応したラジエータ通過流量の比率を決定し、このラジエータ通過流量比率となるように流量制御弁４５を作動制御（弁体開度の連続的制御あるいは弁体開閉のデューティ比制御等）する。そして、冷房時にエンジン４１が停止している場合は、図４の特性図から冷却水のラジエータ通過流量の比率＝１となるように、流量制御弁４５がバイパス通路４７を全閉し、ラジエータ４２への入口流路を全開する状態に操作される。従って、電動ウォータポンプ４４から出た冷却水は、流量制御弁４５を

通過して、その全流量がラジエータ４２に流入する。

【００４０】これにより、冷却水の全流量がラジエータ４２で外気と熱交換して冷却される。この冷却されて低温となった冷却水は水冷媒熱交換器３１の冷却水通路３１ｂに流入し、冷凍サイクルの圧縮機吐出ガスと熱交換を行い、吐出ガスの冷却を行う。水冷媒熱交換器３１で加熱された冷却水は温水ヒータコア４３、エンジン冷却部４１ａを通り電動ウォータポンプ４４に戻る。冷房時にはエアミックスドア１３が温水ヒータコア４３の空気入口部を閉じているため、温水ヒータコア４３と空気との熱交換は行われない。

【００４１】エンジン４１が作動している場合も電磁弁４６が閉じているため、電動ウォータポンプ４４から出た冷却水は流量制御弁４５に流入する。そして、この冷却水は、流量制御弁４５によって図４に示すようにエンジン水温に応じた比率でラジエータ４２を通る冷却水とラジエータ４２をバイパスする冷却水に分流される。ラジエータ４２を通る冷却水は、ラジエータ４２で冷却された後、水冷媒熱交換器３１に流入し、冷凍サイクルの吐出ガス冷媒と熱交換を行い、吐出ガス冷媒の冷却を行う。

【００４２】ラジエータ４２をバイパスして、バイパス通路４７を通過した冷却水は、水冷媒熱交換器３１の出口でラジエータ４２を通った冷却水と合流し、ヒータコア４３、エンジン冷却部４１ａを通り電動ウォータポンプ４４に戻る。この時も、エアミックスドア１３がヒータコア４３の空気入口部を閉じているため、ヒータコア４３で冷却水と空気との熱交換は行われない。

【００４３】以上の作動により、送風機３により送風された空気は蒸発器２３により冷却され、ヒータコア４３を通過しないで車室内に吹き出されるため、車室内の冷房を行うことができる。この時、第１中間圧力冷媒で冷却される車両搭載の発熱部品２５ａの冷却温度が、外気温度、発熱量の変化にかかわらず、常に最適に制御されているので、夏期の高外気温時にも発熱部品２５ａを十分冷却でき、発熱部品２５ａの冷却不足を招くことはない。

【００４４】さらに、冷凍サイクル２０の高圧冷媒は通常の凝縮器２２だけでなく、水冷媒熱交換器３１を介して冷却水に熱を伝え、ラジエータ４２を使って外気に放熱することが可能となるから、発熱部品２５ａの冷却を行い、サイクルの熱負荷が増えても高圧の上昇を抑制でき、消費電力の増加を防止できる。なお、所望の吹出温度による除湿運転を行うときは、エアミックスドア１３を所定角度開いて、ヒータコア４３に所定割合の空気（冷風）を通過させることにより、蒸発器２３で冷却、除湿された冷風の一部を再加熱して、所望温度の冷風を得ることができる。

【００４５】次に、暖房時の作動を説明すると、エンジン４１が長時間停止している場合、またはエンジン４１が作動していても発熱量が小さい場合は冷却水温度が低い。従って、この場合はエンジン４１から流出した冷却水そのままでは冷却水温度が低すぎて暖房能力の不足が生じる。そこで、このような場合には冷凍サイクルを運転して冷却水を加熱し暖房を行う。

【００４６】すなわち、冷凍サイクルの電動式の圧縮機２１を作動させ、圧縮機２１から吐出された吐出ガス冷媒と冷却水とを水冷媒熱交換器３１で熱交換させる。これにより、吐出ガス冷媒を冷却すると同時に冷却水を加熱することができる。ここで、暖房時には、図３のステップ１００からステップ１０６に進み、冷凍サイクル２０の電磁弁３２および電磁弁３０をともに開弁する。

【００４７】電磁弁３２の開弁により、水冷媒熱交換器３１から出た冷媒は、バイパス通路３２ａを通過して凝縮器２２をバイパスする。従って、圧縮機吐出ガス冷媒からの放熱はすべて水冷媒熱交換器３１で行われ、この水冷媒熱交換器３１にて圧縮機吐出ガス冷媒が冷却され、凝縮する。その後、冷媒は冷房時と同様に第１電気膨張弁２４にて第１中間圧力まで減圧され、この第１中間圧力に減圧された冷媒により車両搭載の発熱部品２５ａの冷却を行う。

【００４８】そして、暖房時には、電磁弁３０も開いているため、バイパス通路３０ａを通過して第２電気膨張弁２６にて第２中間圧力まで減圧される。この第２中間圧力の冷媒が気液分離器２７で気液分離され、液冷媒は温度式膨張弁２８により低圧まで減圧され、この低圧冷媒は蒸発器２３を通り、ここで空調ダクト２内の空調空気から吸熱して蒸発する。蒸発後のガス冷媒は吸入ポート２１ｃから圧縮機２１に吸入される。また、気液分離器２７内のガス冷媒はインジェクション通路２１ｆを通りインジェクションポート２１ｄから圧縮機２１に吸入される。

【００４９】一方、冷却水は、電動ウォータポンプ４４から送り出され、図１の冷却水回路４０内を循環するのであるが、その場合、図３のステップ１０７にてエンジン冷却部出口温度センサ６５の検出値が所定値（例えば６０℃）より低いと判定されると、ステップ１０８に進み、電磁弁４６を開く。従って、電動ウォータポンプ４４からの温水はバイパス通路４８を通り、ラジエータ４２をバイパスして水冷媒熱交換器３１に流入する。水冷媒熱交換器３１内で冷凍サイクルの吐出ガス冷媒と冷却水とが熱交換を行い、ガス冷媒を冷却、凝縮するとともに冷却水は加熱される。

【００５０】水冷媒熱交換器３１で加熱された冷却水は直ちにヒータコア４３に流入する。ここで、暖房時にはエアミックスドア１３がバイパス通路４３ａを閉じて、ヒータコア４３の空気入口部を開いているため、送風機３から送られた空気はすべてヒータコア４３を通り、冷却水と熱交換を行って加熱され温風となり、車室内に吹き出され暖房を行う。ヒータコア４３で放熱して低温と

なった冷却水は、エンジン冷却部41aを通り電動ウォータポンプ44に再び戻る。

【0051】ところで、暖房時においてエンジン41が一時的に停止していても、冷却水温度が高い場合は、ステップ107の判定がYESとなり、ステップ108にて電磁弁46を閉じる。従って、ステップ108に進み、流量制御弁45の制御（ラジエータ通過流量の制御）を行う。暖房時のエンジン停止時には、図4の制御マップから冷却水のラジエータ通過流量の比率が0となるように、流量制御弁45がバイパス通路47を全開し、ラジエータ42への入口流路を全閉する状態に操作される。従って、電動ウォータポンプ44から出た冷却水の全流量が、流量制御弁45からバイパス通路47を通過して直接、水冷媒熱交換器31の出口側に流れる。

【0052】従って、冷却水はラジエータ42での放熱および水冷媒熱交換器31での吸熱を行うことなく、ヒータコア43に直接流入し、ここで、空気と熱交換を行い、車室内を暖房する。その後、冷却水はエンジン冷却部41aを通り電動ウォータポンプ44に戻る。暖房時の熱収支について考察すると、冷凍サイクル20は、蒸発器23で送風機3から送られた空気から吸熱するとともに、車両搭載の発熱部品25aの冷却により吸熱を行う。そして、これらの吸熱分に圧縮機21による圧縮仕事分を加えて、水冷媒熱交換器31にて冷却水に放熱を行う。冷却水は、水冷媒熱交換器31で加熱された分とエンジン廃熱がある場合はエンジン廃熱を加えて、ヒータコア43で空気に放熱を行う。

【0053】従って、送風機3から送られた空気は、蒸発器23で冷却された以上に車両搭載の発熱部品25aの廃熱と圧縮仕事分が加えられ加熱されるため、吸い込んだ空気温度以上に加熱されることとなり、車室内の暖房を行うことができる。これにより、エンジン41が長時間停止している場合またはエンジン41が作動しているが発熱量が小さい場合でも、車両搭載の発熱部品25aの冷却熱を有効利用して暖房能力を向上できる。

【0054】また、エンジン41が作動しているため、エンジン発熱量が大きく、冷却水温度が高く、通常のガソリンエンジン車と同様にエンジン廃熱により暖房を行うことができる場合は、エンジン冷却部41aで加熱された冷却水を、電動ウォータポンプ44により送り出すとともに、流量制御弁45によりエンジン水温に応じた比率でラジエータ42を通る冷却水とバイパス通路47を通る冷却水とに分流する。

【0055】ラジエータ42を通る冷却水は、ラジエータ42で冷却された後に水冷媒熱交換器31に流入し、冷凍サイクルの吐出ガス冷媒と熱交換を行い加熱される。また、バイパス通路47を通過してラジエータ42をバイパスした冷却水は、冷却されることなく高温のまま、水冷媒熱交換器31の出口側でラジエータ42を通過した冷却水と合流し、ヒータコア43に入る。

【0056】暖房時は、エアミックスドア13がヒータコア43の空気入口部を開いて、バイパス通路43aを閉じるため、送風機3から送られた空気はヒータコア43を通過するときに、冷却水と熱交換して加熱され、車室内に吹き出され、暖房を行う。ヒータコア43で放熱して低温となった冷却水は、エンジン冷却部41aに戻る。

【0057】一方、冷凍サイクル20はエンジン停止時と同様に作動し、車両搭載発熱部品25aの冷却を行う。これにより、エンジン廃熱が十分にある場合はエンジン廃熱により暖房が行われ、車両搭載の発熱部品25aの冷却も冷凍サイクルにより同時に行われる。以上説明したように、車両搭載の発熱部品25aの冷却は冷凍サイクルの中間圧冷媒により行われるため、外気温度が40℃を越えるような夏期高温時でも外気温度によらず常に必要な発熱部品冷却量を確保できる。

【0058】また、水冷媒熱交換器31を用いて冷媒と冷却水とを熱交換することにより、冷房時の冷凍サイクルの放熱を、凝縮器22だけでなく、水冷媒熱交換器31を通してラジエータ42でも放熱することができる。そのため、サイクル全体としての放熱能力を向上できるので、高圧の上昇を抑制して圧縮機21の消費電力を低減できる。

【0059】さらに、暖房時には車両搭載の発熱部品25aの廃熱を利用して水冷媒熱交換器を通して暖房を行うことが可能となり、エンジン廃熱がない場合でも暖房が可能となる。

【図面の簡単な説明】

【図1】本発明の一実施形態の全体システム構成図である。

【図2】図1の冷凍サイクルのモリエル線図である。

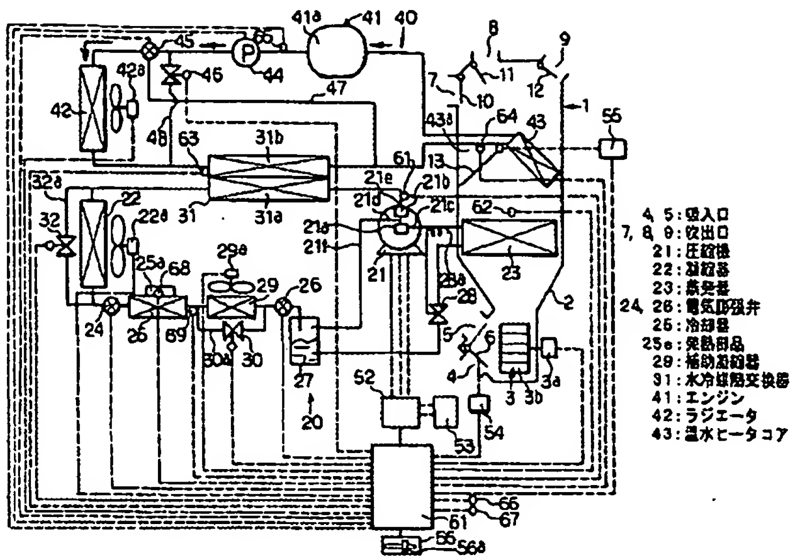
【図3】本発明の一実施形態の作動を説明するフローチャートである。

【図4】図1の冷却水回路に設置される流量制御弁の作用を示す制御特性図である。

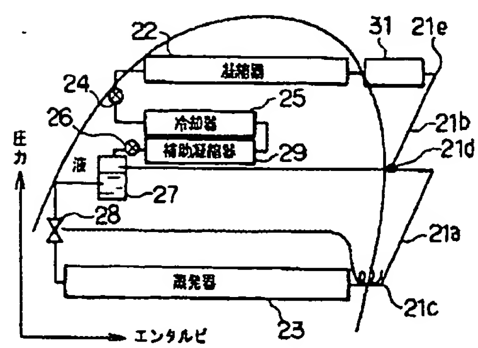
【符号の説明】

2…空調ダクト、3…送風機、4、5…吸入口、7、8、9…吹出口、21…圧縮機、22…凝縮器、23…蒸発器、24…第1電気膨張弁（高圧側減圧手段）、25…冷却器、25a…発熱部品、27…気液分離器、28…温度式膨張弁（低圧側減圧手段）、29…補助凝縮器、31…水冷媒熱交換器、41…エンジン、42…ラジエータ、43…温水ヒータコア。

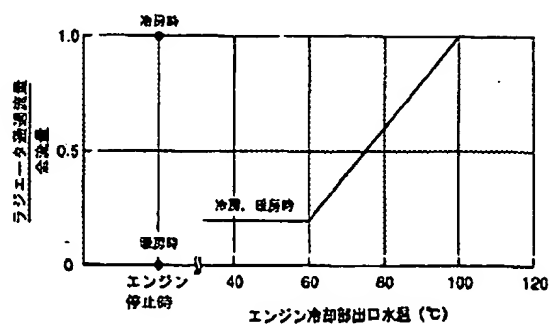
【図1】


4, 5：吸入口
7, 8, 9：吹出口
21：圧縮機
22：凝縮器
23：蒸発器
24, 26：電気膨張弁
25：冷却器
25a：発熱部品
29：補助凝縮器
31：水冷媒熱交換器
41：エンジン
42：ラジエータ
43：温水ヒータコア

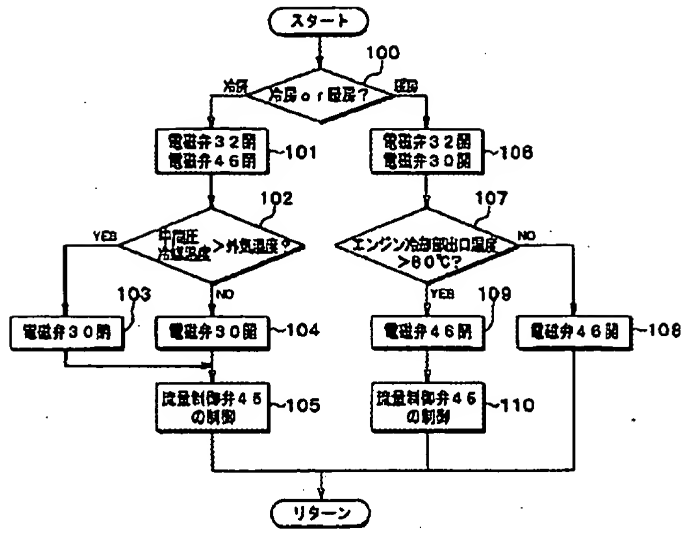
【図2】


凝縮器
冷却器
補助凝縮器
蒸発器
圧力
液
エンタルピ

【図4】


冷房時
暖房時
冷房、暖房時
ラジエータ通過流量/全流量
1.0
0.5
0
エンジン停止時
40
60
80
100
120
エンジン冷却部出口水温（℃）

【図3】



フロントページの続き

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PAT-NO: JP411034640A

DOCUMENT-IDENTIFIER: JP 11034640 A

TITLE: VEHICLE AIR CONDITIONER

PUBN-DATE: February 9, 1999

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APPL-NO: JP09198831

APPL-DATE: July 24, 1997

INT-CL (IPC): B60H001/03, B60H001/22

ABSTRACT:

PROBLEM TO BE SOLVED: To allow for heating by effectively using heat drawn from cooling on-vehicle heating parts in winter, and to allow for cooling on-vehicle heating parts in summer.

SOLUTION: A water coolant heat exchanger 31 is provided on the discharge side of a compressor 21 of a refrigerating cycle 20 for heating cooling water by using gas coolant discharged from the compressor. A condenser 25 is provided to cool an on-vehicle heat producing component 25a by using coolant of an intermediate cycle pressure. An air-conditioning duct 2 is provided with an evaporator 23 and a hot water heater core 43. A cooling water circuit 40 is provided with an engine 41, a radiator 42, and an electric pump 44, in addition to the water coolant heat exchanger 31 and the hot water heater core 43. Heat resulting from cooling the on-vehicle heat producing component 25a pumped up by the compressor 21 is radiated to cooling water, thereby cooling can be performed from the heat producing components when the outside temperature is high.

* NOTICES *

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1.This document has been translated by computer. So the translation may not reflect the original precisely.
2.**** shows the word which can not be translated.
3.In the drawings, any words are not translated.

DETAILED DESCRIPTION

[Detailed Description of the Invention]
[0001]
[Field of the Invention] This invention relates to cooling and a deployment of the heat of cooling (waste heat) of mounted exoergic components especially about an electric vehicle (EV) and the air conditioner for cars suitable as an object for hybrid cars (HV).
[0002]
[Description of the Prior Art] The system which heats by using the heat of cooling of mounted exoergic components effectively for winter as an air conditioner for EV and HV from before is proposed. For example, the air conditioner used for heating is shown in JP,8-258548,A, using the heat of cooling of mounted exoergic components as a heat sink of heat pump. Moreover, the air conditioner which heats by heating further the cooling water warmed by the heat of cooling of mounted exoergic components by heat pump is shown in JP,8-197937,A.
[0003]
[Problem(s) to be Solved by the Invention] However, conventionally [these], with equipment, since a circulating water temperature rises at about 65 degrees C at the time of a summer elevated temperature to which an OAT exceeds 40 degrees C since mounted exoergic components are cooled by cooling water in a summer, it circulates through this cooling water at a radiator and heat is radiated in the open air in the heat of cooling water with the radiator, there is all a problem that mounted exoergic components cannot fully be cooled.
[0004] In view of the point describing above, this invention uses the heat of cooling of mounted exoergic components effectively, makes heating possible, and, on the other hand, aims at offering the air conditioner for cars which sufficiently enables cooling of mounted exoergic components in a summer in winter.
[0005]
[Means for Solving the Problem] In order to attain the above-mentioned purpose, in invention according to claim 1 While preparing the water-cooled intermediation heat exchanger (31) which performs heat exchange between a compressor regurgitation gas refrigerant and cooling water to the compressor (21) discharge side of a refrigerating cycle, and heats this cooling water to it Liquid cooling intermediation of the cycle high-tension side is decompressed to an intermediate pressure with the reduced pressure means (24) of the high-tension side. When the condensator (25) with which the refrigerant of this intermediate pressure flows is formed, and the refrigerant of an intermediate pressure carries out endoergic and evaporates from the exoergic components (25a) of car loading in this condensator (25) Exoergic components (25a) are cooled. To an air-conditioning air duct (2) From air, carry out endoergic [of the refrigerant of the cycle low-tension side], evaporate it, install the evaporator (23) which cools air, and it sets to an air-conditioning air duct (2). The hot water heater core (43) which carries out heat exchange of the air to cooling water, and heats it to the air downstream of an evaporator (23) is installed. Furthermore, the engine for obtaining the driving force of a car directly or indirectly in the cooling water circuit (40) which circulates through cooling water to this hot water heater core (43) (41), It is

characterized by the radiator (42) which carries out heat exchange of the cooling water between the open air, and is cooled, the pump (44) which makes a cooling water circuit (40) circulate through cooling water, and having a water-cooled intermediation heat exchanger (31).
[0006] According to this, while cooling mounted exoergic components (25a) with the intermediate-pressure refrigerant of the refrigerating cycle for car air-conditioning, after pumping up the heat of cooling of mounted exoergic components (25a) with a compressor (21) by using the water-cooled intermediation heat exchanger (31) which carries out heat exchange of the compressor regurgitation refrigerant and cooling water of a refrigerating cycle, heat can be radiated to cooling water. Therefore, even if it is at the high outside-air-temperature time of a summer, mounted exoergic components (25a) can fully be cooled with an intermediate-pressure refrigerant.
[0007] And since heat can be radiated also with a radiator (42) not only through a condenser (22) but through a water-cooled intermediation heat exchanger (31) in heat dissipation of the refrigerating cycle at the time of summer air conditioning, the heat dissipation capacity as the whole cycle can be improved. Therefore, a high-pressure rise is controlled and the consumption power (power consumption) of a compressor (21) can be reduced. Furthermore, at the time of winter heating, cooling water is heated through a water-cooled intermediation heat exchanger (31) using the waste heat of the exoergic components (25a) of car loading, and it becomes possible to heat effectively using this cooling water. Therefore, heating becomes possible even when there is no engine waste heat.
[0008] Moreover, since the refrigerating cycle with a gas injection function is constituted from invention according to claim 2 to invention according to claim 1, effectiveness, such as heating capacity increase by reduction of the compression power by the fall of the compression ratio in the compressor (21) by the gas injection function and the refrigerant circulating load increase to a water-cooled intermediation heat exchanger (31), can be demonstrated. In invention according to claim 3, moreover, to the downstream of the condensator (25) of mounted exoergic components (25a) The bypass path (30a) which bypasses the auxiliary condenser (29) which heat exchange of the refrigerant of an intermediate pressure is carried out [condenser] to the open air, and makes it condense, and this auxiliary condenser (29) is arranged. A detection means to detect the physical quantity which arranges a closing motion valve (30) to this bypass path (30a), and is equivalent to the temperature of the refrigerant of an intermediate pressure (69), It is characterized by having a detection means (66) to detect an OAT, opening a closing motion valve (30), when the temperature of the refrigerant of an intermediate pressure is lower than an OAT, bypassing said auxiliary condenser (29) and pouring the refrigerant of an intermediate pressure.
[0009] According to this, in an auxiliary condenser (29), the refrigerant of an intermediate pressure can carry out endoergic from the open air, and can prevent certainly the fault of worsening cycle efficiency, and, on the other hand, cycle capacity can be increased according to the refrigerant condensation operation by the auxiliary condenser (29) at the time of the conditions which can cool and condense the refrigerant of an intermediate pressure with an auxiliary condenser (29). Moreover, in invention according to claim 4, it is characterized by making it the cooling water cooled with the radiator (42) flow into a water-cooled intermediation heat exchanger (31) at the time of air conditioning operation at least.
[0010] According to this, the regurgitation gas refrigerant in a water-cooled intermediation heat exchanger (31) can be efficiently cooled by the cooling water of the low temperature cooled with the radiator (42). Moreover, in invention according to claim 5, it is characterized by making it the cooling water heated by the water-cooled intermediation heat exchanger (31) flow into a hot water heater core (43) at the time of heating operation at least.
[0011] Since it flows into a hot water heater core (43) immediately, without the cooling water heated by the water-cooled intermediation heat exchanger (31) radiating heat in other parts of a cooling water circuit (40) according to this, heating capacity can be improved effectively. Moreover, the bypass path which bypasses a radiator (42) and pours cooling water in invention according to claim 6 (48), It has a detection means (65) to detect the closing motion valve (46) which opens and closes this bypass path (48), and the circulating water temperature of a cooling water circuit (40). It is characterized by opening a closing motion valve (46), bypassing a radiator (42), and pouring cooling water at the time of heating

operation, when a circulating water temperature is lower than a predetermined value.
[0012] According to this, when a circulating water temperature is low, heat dissipation of the cooling water in a radiator (42) is prevented, and heating capacity can be secured. Moreover, it is characterized by having the flow control valve (45) which controls by invention according to claim 7 the ratio of the cooling water flow rate which passes a radiator (42), and the cooling water flow rate which bypasses a radiator (42) according to a circulating water temperature.
[0013] According to this, the suitable circulating water temperature corresponding to change of the service condition of air conditioning and heating can be set up by controlling the ratio of the cooling water passage flow rate to a radiator (42) by operation of a flow control valve (45). In addition, the sign in the parenthesis given to each above-mentioned means and each means given in a claim shows correspondence relation with the concrete means given in an operation gestalt mentioned later.
[0014]
[Embodiment of the Invention] Hereafter, the operation gestalt which shows this invention in drawing is explained. Drawing 1 shows the whole air-conditioner system configuration for cars by 1 operation gestalt of this invention. Drawing 1 indicates the case where this invention is applied to be an engine (internal combustion engine) 41 to the hybrid car (HV) possessing both motors (not shown) as a car transit driving source. The air-conditioning unit 1 for cars is installed in the instrument panel lower part of the vehicle indoor anterior part of a car, and the air-conditioning duct 2 constitutes the path for air-conditioning which leads air-conditioning air to the vehicle interior of a room.
[0015] The inhalation openings 4 and 5 which inhale inside-and-outside mind are formed in the end side of this air-conditioning duct 2. Change closing motion of the bashful inhalation opening 4 and the open air inhalation opening 5 is carried out by the inside-and-outside mind change door 6. Closing motion control of this inside-and-outside mind change door 6 is carried out by the servo motor 54. The above-mentioned inhalation openings 4 and 5 are adjoined, the blower 3 which ventilates air is installed in the air-conditioning duct 2, and this blower 3 is constituted by centrifugal fan 3b driven by motor 3a and this motor 3a. On the other hand, two or more outlets 7, 8, and 9 which pass to the vehicle interior of a room are formed in the other end side of the air-conditioning duct 2. Change closing motion of these outlets 7, 8, and 9 is carried out by the blow-off mode change doors 10, 11, and 12, respectively, and blow-off modes, such as a face, a bilevel, a foot, and a defroster, are set up.
[0016] Moreover, in the air-conditioning duct 2 in the air downstream, the evaporator 23 of a refrigerating cycle 20 is formed from the blower 3. The low voltage refrigerant of a refrigerating cycle carries out endoergic [of this evaporator 23] from air, and it cools air. Furthermore, the hot water heater core 43 of the cooling water circuit 40 is formed in the air downstream from the evaporator 23, and this hot water heater core 43 heats air by making cooling water (warm water) into a heat source.
[0017] Bypass way 43a is formed in the side of this hot water heater core 43, and the air mix door 13 is arranged in the air inlet section of the hot water heater core 43 free [rotation]. This air mix door 13 adjusts the flow of the air (warm air) which passes the hot water heater core 43, and the flow of the air (cold blast) which passes bypass way 43a and bypasses the hot water heater core 43. As for this air mix door 13, opening (the amount of rotation) is controlled by the servo motor 55.
[0018] Next, the configuration of a refrigerating cycle 20 is explained. The refrigerating cycle is constituted by the following devices other than said evaporator 23. that is, a compressor 21 is driven by the motor (not shown) -- it is electromotive and the water-cooled intermediation heat exchanger 31 is formed in the discharge side of a compressor 21. Refrigerant passage 31a to which the regurgitation gas refrigerant of a compressor 21 flows, and cooling water path 31b to which the cooling water of the below-mentioned cooling water circuit 40 flows are prepared in this water-cooled intermediation heat exchanger 31, and heat exchange is performed to it between the compressor regurgitation gas refrigerant of refrigerant passage 31a, and the cooling water of cooling water path 31b.
[0019] A condenser 22 is formed in the downstream of this water-cooled intermediation heat exchanger 31, the open air and heat exchange are performed, it is cooled, and a gas refrigerant is condensed here. Bypass path 32a is prepared in juxtaposition at this condenser 22, and this bypass path 32a is opened and closed with a solenoid valve 32. The 1st electrical-and-electric-equipment expansion valve (the 1st

reduced pressure means, high-tension-side reduced pressure means) 24 is formed in the downstream of the parallel circuit of this condenser 22 and bypass path 32a, and a high-pressure refrigerant is decompressed to the 1st intermediate pressure by this 1st electrical-and-electric-equipment expansion valve 24. As for the 1st electrical-and-electric-equipment expansion valve 24, whenever [valve-opening] (the amount of diaphragms) is electrically controlled so that the 1st intermediate pressure turns into target preassure force.

[0020] And a condensator 25 is formed in the downstream of the 1st electrical-and-electric-equipment expansion valve 24, and this condensator 25 cools exoergic components 25a of car loading with the 1st intermediate-pressure refrigerant. As exoergic components 25a of car loading here, they are the drive motor of HV and EV car and the solid state switch component (power transistor) of the inverter for revolving speed control, a mounted dc-battery, etc., for example.

[0021] The auxiliary condenser 29 is formed in the downstream of a condensator 25, and this auxiliary condenser 29 condenses the 1st intermediate pressure refrigerant by performing heat exchange between the 1st intermediate pressure refrigerant and the open air. Bypass path 30a is prepared in juxtaposition also at this auxiliary condenser 29, and this bypass path 30a is opened and closed with a solenoid valve 30. The 2nd electrical-and-electric-equipment expansion valve (the 2nd reduced pressure means) 26 is formed in the downstream of the parallel circuit of this auxiliary condenser 29 and a solenoid valve 30, and this 2nd electrical-and-electric-equipment expansion valve 26 decompresses the 1st intermediate-pressure refrigerant to the 2nd intermediate pressure further. As for this 2nd electrical-and-electric-equipment expansion valve 26, whenever [valve-opening] (the amount of diaphragms) is electrically controlled so that the 2nd intermediate pressure turns into target preassure force.

[0022] The vapor-liquid-separation machine 27 is formed in the downstream of the 2nd electrical-and-electric-equipment expansion valve 26, and this vapor-liquid-separation machine 27 achieves the function to accumulate liquid cooling intermediation while performing vapor liquid separation of the refrigerant of the 2nd intermediate pressure. The liquid cooling intermediation by which vapor liquid separation was carried out with this vapor-liquid-separation vessel 27 is decompressed by the temperature type expansion valve (the 3rd reduced pressure means, low-tension side reduced pressure means) 28 to a low voltage pressure. This temperature type expansion valve 28 has temperature sensor barrel 28a which senses the temperature of the outlet refrigerant (compressor inhalation refrigerant) of an evaporator 23, and it adjusts whenever [valve-opening] (refrigerant flow rate) so that the degree of superheat of an evaporator outlet refrigerant may be maintained by the predetermined value.

[0023] A compressor 21 is a gas injection mold compressor which consists of high rank side compression zone 21b which compresses the mixed gas of low stage side compression zone 21a (refer to the Mollier chart of drawing 2) which inhales the refrigerant of the refrigerating cycle low-tension side from inhalation port 21c, and is compressed to an intermediate pressure, the gas refrigerant compressed to the intermediate pressure, and the gas refrigerant which flows from gas injection port 21d to a discharge pressure, and carries out the regurgitation from regurgitation port 21e, for example, consists of a scrolling compressor.

[0024] Moreover, alternating voltage is impressed to the motor of the refrigerant compressor 21 by the inverter 52, and motor rotational speed is continuously changed by adjusting the frequency of alternating voltage with this inverter 52. The direct current voltage from the mounted dc-battery 53 is impressed to this inverter 52. On the other hand, cooling section 41a of the engine (internal combustion engine) 41 which serves as a driving source of a car directly and indirectly is prepared in the cooling water circuit 40. Here, an engine 41 may use the engine 41 else [in the case of taking out the power directly and mechanically and using as a transit driving source of a car] as only for generator drives. In this case, drive of a drive motor and charge to the mounted dc-battery 53 are performed with the generation-of-electrical-energy output of a generator, and it runs a car.

[0025] In the cooling water circuit 40, the closed circuit where cooling water passage 31b of the cooled radiator 42 which carries out heat exchange to the open air, and the above mentioned water-cooled intermediation heat exchanger 31, and said hot water heater core 43 are arranged at an engine 41 and a serial, and cooling water circulates through electric Water pump 44 and cooling water which can be

driven is constituted, using the mounted dc-battery 53 as a power source also at the time of a halt of an engine 41. Electric Water pump 44 feeds cooling water in the direction (the direction of an arrow head) which faces to a radiator 42 from an engine 41.

[0026] Moreover, while forming the 1st bypass path 47 which bypasses cooling water passage 31b of a radiator 42 and the water-cooled intermediation heat exchanger 31, the 2nd bypass path 48 which bypasses a radiator 42 and a flow control valve 45 is formed. The flow control valve 45 is formed in the branch point of the inlet-port section of a radiator 42, and the inlet-port section of the 1st bypass path 47. the ratio of the circulating water flow which this flow control valve 45 is a cross valve type thing, and radiates heat with a radiator 42, and the circulating water flow which bypasses a radiator 42 -- being continuous (linear) -- adjustable -- it is controllable. For this reason, the flow control valve 45 is electrically constituted possible [adjustment] in whenever [valve-opening]. On the other hand, the solenoid valve 46 which opens and closes a path is formed in the 2nd bypass path 48.

[0027] By the way, in order to carry out energization control of the electrical machinery and apparatus of the above-mentioned flow control valve 45 and solenoid-valve 46 grade, it has the air-conditioning control unit 51, and this control unit 51 for air-conditioning is a microcomputer and an electronic control which consists of that circumference circuit. The control device 51 for air-conditioning also performs energization control of an inverter 52, the servo motor 54 for inside-and-outside mind doors, the servo motor 55 for air mix doors, electric Water pump 44, a blower 3, the open air blower fans 22a, 29a, and 42a, etc. besides the above-mentioned valves.

[0028] An input signal is applied to the control unit 51 for air-conditioning from the various following sensors etc. that is The temperature of a compressor regurgitation gas refrigerant The discharge-temperature sensor 61 to detect, the after [an evaporator] temperature sensor 62 which detects the air temperature immediately after blow off of an evaporator 23, the water-cooled intermediation inlet temperature sensor 63 which detects the cooling water inlet temperature of the water-cooled intermediation heat exchanger 31, the heater core inlet-port water temperature sensor 64 which detects the inlet-port water temperature of the hot water heater core 43, The cold water temperature of engine-coolant section 41a The engine water temperature sensor 65 and OAT to detect The outside-air-temperature sensor 66 to detect and the air temperature of the vehicle interior of a room Various levers, such as temperature control lever 56a of the bashful temperature sensor 67 to detect, the cooling temperature sensor 68 which detects the cooling temperature of exoergic components 25a, the intermediate pressure refrigerant temperature sensor 69 which detects the coolant temperature of the exoergic components condensator 25, and air conditioning controls 56, The air-conditioning actuation signal from a switch is inputted into the control unit 51 for air-conditioning.

[0029] Next, actuation is explained in the above-mentioned configuration. It judges whether drawing 3 is in whether an air conditioner is in air conditioning operation, and heating operation in step 100 by showing the control routine by the control device 51 for air-conditioning, and a control routine starting by injection of the air-conditioning actuation switch (not shown) of air conditioning controls 56. This judgment is performed by whether the actuated valve position of temperature control lever 56a of air conditioning controls 56 is in a low temperature side (air conditioning side), or it is in an elevated-temperature side (heating side).

[0030] If air conditioning operation is judged in step 100, it will progress to step 101 and the solenoid valve 32 of a refrigerating cycle 20 and the solenoid valve 46 of the cooling water circuit 40 will be closed. While the engine 41 is operating on the usual load conditions at the time of air conditioning operation, the circulating water temperature which flows into the water-cooled intermediation heat exchanger 31 with the heat dissipation engine performance of a radiator 42 is near 60 degree C of outlines, and, on the other hand, the gas coolant temperature breathed out from a compressor 21 is usually near 80-degreeC.

[0031] Therefore, in the water-cooled intermediation heat exchanger 31, since a circulating water temperature is that it is lower than a coolant temperature and ****, by the water-cooled intermediation heat exchanger 31, the gas refrigerant of elevated-temperature high pressure breathed out from the compressor 21 radiates heat to cooling water, and is cooled. Consequently, some compressor

regurgitation gas refrigerants are condensed in refrigerant path 31a of the water-cooled intermediation heat exchanger 31. Here, since the solenoid valve 32 of a refrigerating cycle has closed at step 101 like **** at the time of air conditioning, with a condenser 22, the refrigerant which came out of refrigerant path 31a of the water-cooled intermediation heat exchanger 31 performs the open air and heat exchange, and it is cooled and it condenses them.
[0032] This elevated temperature and high-pressure liquid cooling intermediation are decompressed to the 1st intermediate pressure by the 1st electrical-and-electric-equipment expansion valve 24 next, and will be in a vapor-liquid two phase condition. And this vapor-liquid two phase refrigerant flows into a condensator 25, and cools exoergic components 25a of car loading. That is, in a condensator 25, it cools exoergic components 25a while endoergic [of the liquid cooling intermediation in a vapor-liquid two phase refrigerant] is carried out and it evaporates from exoergic components 25a of car loading.
[0033] Here, according to the calorific value of exoergic components 25a, the 1st intermediate-pressure coolant temperature is controlled by the 1st electrical-and-electric-equipment expansion valve 24 so that the cooling temperature of exoergic components 25a detected by the cooling temperature sensor 68 serves as a predetermined value (for example, OAT of +5 degrees C). That is, since it is necessary to increase the amount of cooling when the calorific value of exoergic components 25a is large, opening of the 1st electrical-and-electric-equipment expansion valve 24 is made small, and the 1st (enlarging amount of diaphragms) intermediate pressure (coolant temperature) is made low.
[0034] On the other hand, when the calorific value of exoergic components 25a is small, since there may be few amounts of cooling, they enlarge opening of the 1st electrical-and-electric-equipment expansion valve 24, and make high the 1st (making amount of diaphragms small) intermediate pressure (coolant temperature). Thereby, even if the calorific value of exoergic components 25a of car loading and an OAT change, only a complement can always cool exoergic components 25a appropriately.
[0035] On the other hand, when the coolant temperature detected with the intermediate pressure refrigerant temperature sensor 69 is higher than an OAT, the judgment of step 102 of drawing 3 serves as YES, and progresses to step 103, and a solenoid valve 30 is closed. Therefore, it flows into the auxiliary condenser 29, and the open air and heat exchange are performed, it is cooled again, and the refrigerant which came out of the exoergic components condensator 25 is condensed. In contrast, when the coolant temperature of the 1st intermediate pressure is lower than an OAT, it progresses to step 104 from step 102, and a solenoid valve 30 is opened. Therefore, the 1st intermediate-pressure refrigerant passes bypass path 30a, bypasses the auxiliary condenser 29, and flows into the 2nd electrical-and-electric-equipment expansion valve 26. Thus, when the coolant temperature of the 1st intermediate pressure is lower than an OAT, and the 1st intermediate-pressure refrigerant passes through the bypass path 30 and bypasses the auxiliary condenser 29, the 1st intermediate-pressure refrigerant can prevent beforehand the fault of carrying out endoergic from the open air.
[0036] And by the 2nd electrical-and-electric-equipment expansion valve 26, the 1st intermediate-pressure refrigerant flows into the vapor-liquid-separation machine 27, after being decompressed to the 2nd intermediate pressure. A two phase refrigerant is separated into gas and liquid by the vapor-liquid-separation machine 27, and a gas refrigerant passes along 21f of gas injection piping, and is inhaled in the middle of [d / gas injection port 21] the compression process of a compressor 21. On the other hand, the liquid cooling intermediation in the vapor-liquid-separation machine 27 is decompressed to a low voltage pressure by the temperature type expansion valve 28, and flows into an evaporator 23. In addition, the refrigerant flow rate is controlled by the temperature type expansion valve 28 so that a refrigerant evaporates completely at the outlet of an evaporator 23.
[0037] In the air-conditioning unit 1, with an evaporator 23, a refrigerant and heat exchange are performed, it is cooled, and the air ventilated by the blower 3 performs air conditioning of the vehicle interior of a room. Here, with an evaporator 23, a refrigerant evaporates, and is gasified, and the gas refrigerant is inhaled and compressed into a compressor 21 from inhalation port 21c. The Mollier chart of drawing 2 shows the condition of the refrigerant in each device of the above-mentioned refrigerating cycle.
[0038] Next, when actuation of the cooling water circuit 40 at the time of air conditioning is explained

based on drawing 4 , the axis of abscissa of drawing 4 takes the cold water temperature of engine-coolant section 41a detected by the engine water temperature sensor 65, and an axis of ordinate is the ratio (full flow to a radiator passage flow rate / flow control valve 45) of the radiator passage flow rate controlled by the flow control valve 45. Since the control characteristic (map) of this drawing 4 is set up beforehand, is memorized by ROM of the microcomputer of the control unit 51 for air-conditioning and the solenoid valve 46 of the radiator bypass path 48 has closed it at the time of air conditioning, it can control a radiator passage flow rate by the flow control valve 45.
[0039] Then, in step 105 of drawing 3 , according to the control characteristic of drawing 4 , the ratio of the radiator passage flow rate corresponding to the cold water temperature of engine-coolant section 41a is determined, and actuation control of the flow control valve 45 is carried out so that it may become this rate of radiator passage flow rate (continuous control of valve element opening, or duty ratio control of valve element closing motion). And when the engine 41 has stopped at the time of air conditioning, a flow control valve 45 carries out the close by-pass bulb completely of the bypass path 47, and is operated by the condition of opening the inlet-port passage to a radiator 42 fully so that it may be set to ratio =1 of the radiator passage flow rate of cooling water from the property Fig. of drawing 4 . Therefore, the cooling water which came out of electric Water pump 44 passes a flow control valve 45, and the full flow flows into a radiator 42.
[0040] Thereby, with a radiator 42, the full flow of cooling water carries out heat exchange to the open air, and is cooled. This cooling water that was cooled and became low temperature flows into cooling water path 31b of the water-cooled intermediation heat exchanger 31, performs the compressor regurgitation gas and heat exchange of a refrigerating cycle, and cools regurgitation gas. The cooling water heated by the water-cooled intermediation heat exchanger 31 returns to electric Water pump 44 through the hot water heater core 43 and engine-coolant section 41a. Since the air mix door 13 has closed the air inlet section of the hot water heater core 43 at the time of air conditioning, heat exchange of the hot water heater core 43 and air is not performed.
[0041] Since the solenoid valve 46 has closed also when the engine 41 is operating, the cooling water which came out of electric Water pump 44 flows into a flow control valve 45. And this cooling water is shunted toward the cooling water which passes along a radiator 42 by the ratio according to engine water temperature as a flow control valve 45 shows to drawing 4 , and the cooling water which bypasses a radiator 42. After being cooled with a radiator 42, the cooling water which passes along a radiator 42 flows into the water-cooled intermediation heat exchanger 31, performs the regurgitation gas refrigerant and heat exchange of a refrigerating cycle, and cools a regurgitation gas refrigerant.
[0042] The cooling water which bypassed the radiator 42 and passed through the bypass path 47 joins the cooling water which passed along the radiator 42 by the outlet of the water-cooled intermediation heat exchanger 31, and returns to electric Water pump 44 through the heater core 43 and engine-coolant section 41a. Since the air mix door 13 has closed the air inlet section of the heater core 43 also at this time, heat exchange of cooling water and air is not performed with the heater core 43.
[0043] It is cooled with an evaporator 23, and since the air ventilated with the blower 3 by the above actuation blows off to the vehicle interior of a room without passing the heater core 43, it can perform air conditioning of the vehicle interior of a room. Since the cooling temperature of exoergic components 25a of car loading cooled with the 1st intermediate-pressure refrigerant at this time is controlled always the optimal irrespective of change of an OAT and calorific value, it can cool exoergic components 25a enough also at the time of the high outside air temperature of a summer, and does not cause the lack of cooling of exoergic components 25a.
[0044] Furthermore, the high-pressure refrigerant of a refrigerating cycle 20 conducts heat to cooling water not only through the usual condenser 22 but through the water-cooled intermediation heat exchanger 31, since it becomes possible [radiating heat in the open air using a radiator 42], it can cool exoergic components 25a, even if its thermal load of a cycle increases, can control a high-pressure rise, and can prevent the increment in power consumption. In addition, when performing dehumidification operation by whenever [desired blow-off temperature], by opening the air mix door 13 the degree of predetermined angle, and making the heater core 43 pass the air (cold blast) of a predetermined rate, a

part of cold blast cooled and dehumidified with the evaporator 23 is reheated, and the cold blast of request temperature can be obtained.
[0045] Next, even if the engine 41 is operating, when calorific value is small, a circulating water temperature is low, when the engine 41 is carrying out a long duration halt, if the actuation at the time of heating is explained. therefore, the cooling water which flowed out of the engine 41 in this case -- if it remains as it is, a circulating water temperature is too low and lack of heating capacity arises. Then, it heats by operating a refrigerating cycle in such a case and heating cooling water.
[0046] That is, the electromotive compressor 21 of a refrigerating cycle is operated and heat exchange of the regurgitation gas refrigerant and cooling water which were breathed out from the compressor 21 is carried out by the water-cooled intermediation heat exchanger 31. Thereby, cooling water can be heated while cooling a regurgitation gas refrigerant. Here, at the time of heating, it progresses to step 106 from step 100 of drawing 3 , and both the solenoid valves 32 and solenoid valves 30 of a refrigerating cycle 20 are opened.
[0047] By valve opening of a solenoid valve 32, the refrigerant which came out of the water-cooled intermediation heat exchanger 31 passes bypass path 32a, and bypasses a condenser 22. Therefore, all heat dissipation from a compressor regurgitation gas refrigerant is performed by the water-cooled intermediation heat exchanger 31, a compressor regurgitation gas refrigerant is cooled by this water-cooled intermediation heat exchanger 31, and it condenses. Then, a refrigerant cools exoergic components 25a of car loading with the refrigerant which was decompressed to the 1st intermediate pressure by the 1st electrical-and-electric-equipment expansion valve 24 like the time of air conditioning, and was decompressed by this 1st intermediate pressure.
[0048] And at the time of heating, since the solenoid valve 30 is also opened, bypass path 30a is passed and it decompresses to the 2nd intermediate pressure by the 2nd electrical-and-electric-equipment expansion valve 26. Vapor liquid separation of the refrigerant of this 2nd intermediate pressure is carried out with the vapor-liquid-separation vessel 27, liquid cooling intermediation is decompressed by the temperature type expansion valve 28 to low voltage, it passes along an evaporator 23, and from the air-conditioning air in the air-conditioning duct 2, endoergic [of this low voltage refrigerant] is carried out, and it evaporates here. The gas refrigerant after evaporation is inhaled by the compressor 21 from inhalation port 21c. Moreover, the gas refrigerant in the vapor-liquid-separation machine 27 is inhaled by the compressor 21 from injection port 21d through 21f of injection paths.
[0049] On the other hand, although cooling water is sent out from electric Water pump 44 and circulates through the inside of the cooling water circuit 40 of drawing 1 , if it is judged with the detection value of the engine-coolant section outlet temperature sensor 65 being lower than a predetermined value (for example, 60 degrees C) at step 107 of drawing 3 in that case, it will progress to step 108 and will open a solenoid valve 46. Therefore, the warm water from electric Water pump 44 passes along the bypass path 48, bypasses a radiator 42, and flows into the water-cooled intermediation heat exchanger 31. The regurgitation gas refrigerant and cooling water of a refrigerating cycle perform heat exchange within the water-cooled intermediation heat exchanger 31, and cooling water is heated while cooling and condensing a gas refrigerant.
[0050] The cooling water heated by the water-cooled intermediation heat exchanger 31 flows into the heater core 43 immediately. Since the air mix door 13 closes bypass path 43a at the time of heating and the air inlet section of the heater core 43 is opened here, all the air sent from the blower 3 passes along the heater core 43, it performs cooling water and heat exchange, is heated, serves as warm air, and heats by blowing off to the vehicle interior of a room. The cooling water which radiated heat with the heater core 43 and became low temperature returns to electric Water pump 44 again through engine-coolant section 41a.
[0051] By the way, even if ENSHIN 41 has stopped temporarily at the time of heating, when a circulating water temperature is high, the judgment of step 107 serves as YES and closes a solenoid valve 46 at step 108. Therefore, it progresses to step 108 and a flow control valve 45 is controlled (control of a radiator passage flow rate). A flow control valve 45 opens the bypass path 47 fully, and is operated by the condition of carrying out the close by-pass bulb completely of the inlet-port passage to a

radiator 42 so that the ratio of the radiator passage flow rate of cooling water may be set to 0 from the control map of drawing 4 at the time of the engine shutdown at the time of heating. Therefore, the full flow of the cooling water which came out of electric Water pump 44 passes through the bypass path 47 from a flow control valve 45, and flows to the outlet side of the water-cooled intermediation heat exchanger 31 directly.

[0052] Therefore, without performing heat dissipation with a radiator 42, and endoergic [in the water-cooled intermediation heat exchanger 31], cooling water flows into the heater core 43 directly, performs air and heat exchange and heats the vehicle interior of a room here. Then, cooling water returns to electric Water pump 44 through engine-coolant section 41a. If the heat balance at the time of heating is considered, it will perform endoergic by cooling of exoergic components 25a of car loading while carrying out endoergic [of the refrigerating cycle 20] from the air sent from the blower 3 with the evaporator 23. And a part for the work of compression by the compressor 21 is added to a part for endoergic [these], and heat is radiated to cooling water by the water-cooled intermediation heat exchanger 31. Cooling water adds engine waste heat, when there are the part and engine waste heat which were heated by the water-cooled intermediation heat exchanger 31, and it radiates heat to air with the heater core 43.

[0053] Therefore, since a part for the waste heat of exoergic components 25a of car loading and work of compression is added and heated more than it was cooled with the evaporator 23, the air sent from the blower 3 will be heated more than the absorbed air temperature, and can perform heating of the vehicle interior of a room. By this, when the engine 41 is carrying out a long duration halt, the engine 41 is operating, but even when calorific value is small, the heat of cooling of exoergic components 25a of car loading is used effectively, and heating capacity can be improved.

[0054] Moreover, splitting is carried out to the cooling water passing through the cooling water which passes along a radiator 42 by the ratio [flow control valve / 45] corresponding to engine water temperature while sending out the cooling water heated by engine-coolant section 41a when engine calorific value is large since the engine 41 is operating, a circulating water temperature was high and engine waste heat was able to heat like the usual gasoline engine vehicle with electric Water pump 44, and the bypass path 47.

[0055] The cooling water which passes along a radiator 42 flows into the water-cooled intermediation heat exchanger 31, after being cooled with a radiator 42, performs the regurgitation gas refrigerant and heat exchange of a refrigerating cycle, and is heated. Moreover, without being cooled, in the hot state, the cooling water which passed through the bypass path 47 and bypassed the radiator 42 joins the cooling water which passed the radiator 42 by the outlet side of the water-cooled intermediation heat exchanger 31, and goes into the heater core 43.

[0056] In order for the air mix door 13 to open the air inlet section of the heater core 43 and to close bypass path 43a at the time of heating, when passing the heater core 43, heat exchange of it is carried out to cooling water, and the air sent from the blower 3 is heated, and heats by blowing off to the vehicle interior of a room. The cooling water which radiated heat with the heater core 43 and became low temperature returns to engine-coolant section 41a.

[0057] On the other hand, a refrigerating cycle 20 operates like the time of an engine shutdown, and cools car loading exoergic components 25a. Thereby, in a certain case, heating is fully performed for engine waste heat by engine waste heat, and cooling of exoergic components 25a of car loading is also performed to coincidence by the refrigerating cycle. Since cooling of exoergic components 25a of car loading is performed by the intermediate pressure refrigerant of a refrigerating cycle as explained above, also in the time of a summer elevated temperature to which an OAT exceeds 40 degrees C, it is not based on an OAT but the always required amount of exoergic components cooling can be secured.

[0058] Moreover, heat can be radiated also with a radiator 42 in heat dissipation of the refrigerating cycle at the time of air conditioning not only through the condenser 22 but through the water-cooled intermediation heat exchanger 31 by carrying out heat exchange of a refrigerant and the cooling water using the water-cooled intermediation heat exchanger 31. Therefore, **** which can improve the heat dissipation capacity as the whole cycle, and a high-pressure rise are controlled, and the power

consumption of a compressor 21 can be reduced.
[0059] Furthermore, heating becomes possible, even when it becomes possible to heat through a water-cooled intermediation heat exchanger using the waste heat of exoergic components 25a of car loading at the time of heating and there is no engine waste heat.

[Translation done.]

DERWENT-ACC-NO: 1999-107488

DERWENT-WEEK: 200025

TITLE: Air conditioning system for vehicle - has cooling unit for heat source coupled to cooling system to remove heat in Summer and to reclaim heat in Winter

INVENTOR: BANZAI, K; ISHII, K ; KOBUBO, A ; SUZUKI, T ; KOKUBO, A

PATENT-ASSIGNEE: DENSO CORP[NPDE] , NIPPONDENSO CO LTD[NPDE]

PRIORITY-DATA: 1997JP-0198831 (July 24, 1997)

PATENT-FAMILY:

PUB-NO	PUB-DATE	LANGUAGE	PAGES	MAIN-IPC
DE 19833251 A1	January 28, 1999	N/A	012	B60H 001/00
US 6047770 A	April 11, 2000	N/A	000	B60H 003/00
JP 11034640 A	February 9, 1999	N/A	010	B60H 001/03

APPLICATION-DATA:

PUB-NO	APPL-DESCRIPTOR	APPL-NO	APPL-DATE
DE 19833251A1	N/A	1998DE-1033251	July 23, 1998
US 6047770A	N/A	1998US-0120911	July 22, 1998
JP 11034640A	N/A	1997JP-0198831	July 24, 1997

INT-CL (IPC): B60H001/00, B60H001/03 , B60H001/22 , B60H001/32 , B60H003/00

ABSTRACTED-PUB-NO: DE 19833251A

BASIC-ABSTRACT:

The air conditioning system has a main air duct with a blower and a heating or cooling block (23) to heat or cool the air into the vehicle. The system also cools an electric unit (25a) and transfers the heat to a coolant circuit to be either dumped via a radiator (22), e.g. in the Summer setting, or to add heat to the coolant for warming the air flow in Winter.

The system is especially applicable in hybrid drive vehicles. It is especially used to cool the regulating circuit of the electric system.

USE - Air-conditioning system for use in road vehicle.

ADVANTAGE - Cost effective, energy efficient air conditioning system.

ABSTRACTED-PUB-NO: US 6047770A

EQUIVALENT-ABSTRACTS:

The air conditioning system has a main air duct with a blower and a heating or cooling block (23) to heat or cool the air into the vehicle. The system also cools an electric unit (25a) and transfers the heat to a coolant circuit to be either dumped via a radiator (22), e.g. in the Summer setting, or to add heat to the coolant for warming the air flow in Winter.

The system is especially applicable in hybrid drive vehicles. It is especially used to cool the regulating circuit of the electric system.

USE - Air-conditioning system for use in road vehicle.

ADVANTAGE - Cost effective, energy efficient air conditioning system.

CHOSEN-DRAWING: Dwg.1/4

TITLE-TERMS: AIR CONDITION SYSTEM VEHICLE COOLING UNIT HEAT SOURCE COUPLE COOLING SYSTEM REMOVE HEAT SUMMER RECLAIM HEAT WINTER

DERWENT-CLASS: Q12 X22

EPI-CODES: X22-J02E;

SECONDARY-ACC-NO:
Non-CPI Secondary Accession Numbers: N1999-077694

copy of JP 59-24134
with Translation
for Appellant.

62-236.6 AU 344 48402

JA 0024134
FEB 1984

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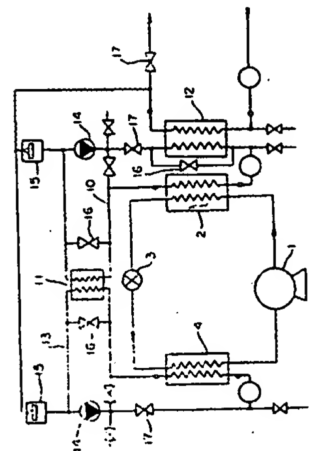
with translation

(54) CALORIMETER FOR AIR CONDITIONER
(11) 59-24134 (A) (43) 7.2.1984 (19) JP
(21) Appl. No. 57-134257 (22) 31.7.1982
(71) AISHIN SEIKI K.K. (72) YUTAKA MOMOSE
(51) Int. Cl³. F24F11/02,G01K17/00

PURPOSE: To miniaturize the meter and facilitate the operation thereof by a method wherein an intermediate heat exchanger, exchanging the dissipating heat of a condensing side and the absorbing heat of an evaporation side, and a device for cooling the difference between heat dissipating amount and heat absorbing amount are provided in this meter.

CONSTITUTION: Freon gas from a compressor 1 is condensed in a condenser 2 and dissipates heat. The dissipated heat is transmitted from a cooling water circuit 10 to a warm-water circuit 13 through the intermediate heat exchanger 11. The freon gas, passed through an expansion valve 3, is evaporated by an evaporator 4, however, heat, absorbed in this case, is supplied by the dissipated heat of the condensing side, which is obtained by the intermediate heat exchanger 11. On the other hand, the heat dissipating amount becomes larger as compared with the heat absorbing amount by an amount corresponding to the power of the compressor 1, therefore, cooling water, passed through the intermediate heat exchanger 11, is cooled by a cooling heat exchanger 12 so as to absorb the amount of heat, which is corresponding to the difference. This cooling may be fulfilled by utilizing city water and a cooling tower is not necessary because the amount of heat in this cooling is small. Calorific amount may be obtained by measuring the temperatures of the cooling water at the outlet and inlet ports of the condenser 2 and the cooling heat exchanger 12 and the temperatures of the warm-water at the inlet and outlet ports of the evaporator 4.



⑲ 日本国特許庁 (JP)　　⑪特許出願公開

⑫ 公開特許公報 (A)　　昭59—24134

⑤Int. Cl.³	識別記号	庁内整理番号	⑬公開 昭和59年(1984)2月7日
F 24 F 11/02		L 7914—3L	
G 01 K 17/00		7269—2F	発明の数 1
			審査請求 未請求

(全 2 頁)

⑤空調器用カロリーメーター

㉑特　　願 昭57—134257

㉒出　　願 昭57(1982)7月31日

⑫発 明 者 百瀬豊

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明　細　書

1 発明の名称

空調器用カロリーメーター

2 特許請求の範囲

蒸発側熱交換器の温水回路と凝縮側熱交換器の冷水回路を熱交換するための中間熱交換器を有し、さらに、前記冷水回路に市水と熱交換する冷却熱交換器を配していることを特徴とする空調器用カロリーメーター。

3 発明の詳細な説明

この発明は冷暖房能力を測定する空調器用カロリーメーターに関する。

冷暖房能力を測定する従来の空調器用カロリーメーターは、第1図に示すように、圧縮器1、凝縮器2、膨脹弁3および蒸発器4を有する空調システムに、凝縮放熱側に冷却塔5を有する冷水回路6を配し、凝縮器2で熱交換を行ない、又、蒸発吸熱側にヒーター7を内蔵した温水タンク8を有する温水回路9を配し、蒸発器4で熱交換を成し、冷水回路6の凝縮器2での熱交換の前後の温度および温水回路9の蒸発器4での熱交換の前後の温度を測定してカロリー計算を行なう。このような従来のカロリーメーターは、冷却水を作る冷却塔5およびヒーター7を内蔵した温水タンク8が必要であり、空調システムの容量によつては、これら冷却塔および温水タンクが大型となり、消費されるエネルギーも大で不経済である。

それ故、この発明は、凝縮側放熱を蒸発側吸熱に利用することにより前述した不経済を解消させるもので、この発明によれば、凝縮側放熱分と蒸発側吸熱分の熱を交換する中間熱交換器と、該放熱量と吸熱量の差分を冷却する装置とを有する技術手段を用いる。

この発明によれば、カロリーメーターは小型となり操作も容易となる；又、放熱量と吸熱量の差分だけ冷却すればよいので、冷却機構も簡単となる。

この発明の実施例を添付第2図を参照して

説明する。

圧縮器1からのクレオンガスは凝縮器2、膨脹弁3および蒸発器4を通つて圧縮器1に還流する。凝縮放熱側の冷水回路10は、凝縮器2での熱交換部と、中間熱交換器11および冷却熱交換器12を有する。又、蒸発吸熱側の温水回路13は、蒸発器での熱交換部と、中間熱交換器11とを有する。冷水および温水回路10，13には、夫々、ポンプ14および補助タンク15を設ける。又、中間熱交換器11の部分にバイパスバルブ16を、そして、冷却熱交換器12の部分にバイパスバルブ16を設ける。17は、流量調整バルブである。

圧縮器1からのフレオンガスは、凝縮器2で凝縮し、放熱する。この放熱は、冷却水回路10により中間熱交換器11で温水回路13に伝達される。膨脹弁3を通つたフレオンガスは蒸発器4で蒸発するが、この際の吸熱は、中間熱交換器11で得た凝縮側放熱によつて賄なわれる。一方、放熱量は吸熱量に比し、圧縮機1の動力分だけ大となるので、中間熱交換器11を通つた冷却水は、冷却熱交換器12で、この差分の熱量を奪うよう冷却される。この冷却は、熱量が小さいので、市水の利用によつて充分達成でき、冷却塔は不用である。カロリーは、凝縮器2および冷却熱交換器12の出入口の冷却水温度、蒸発器4の出入口の温水温度を測定すればよい。

4 図面の簡単な説明

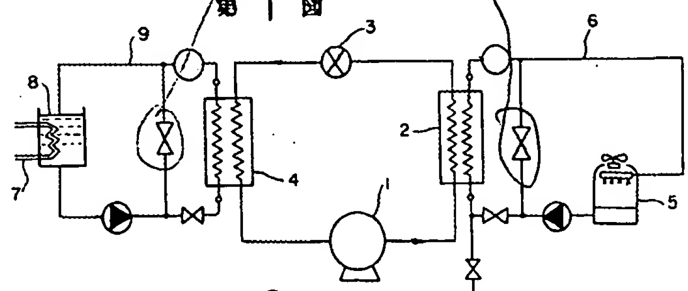
第1図は従来例のカロリーメーターを示す図、第2図はこの発明の一例のカロリーメーターを示す図である。

図中：1…圧縮器、2…凝縮器、3…膨脹弁、4…蒸発器、10…冷水回路、11…中間熱交換器、12…冷却熱交換器、13…温水回路。

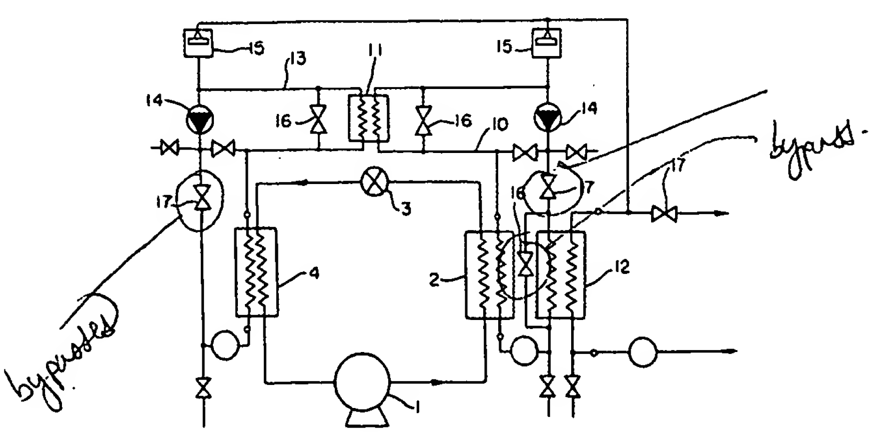
代理人 弁理士 桑 原 英 明

bypasses

第1図



第2図



PTO 05-4656

Japanese Kokai Patent Application
No. Sho 59[1984]-24134

CALORIMETER FOR AIR CONDITIONER

Yukata Momose

UNITED STATES PATENT AND TRADEMARK OFFICE
WASHINGTON, D.C. JULY 2005
TRANSLATED BY THE RALPH MCELROY TRANSLATION COMPANY

JAPANESE PATENT OFFICE
PATENT JOURNAL (A)
KOKAI PATENT APPLICATION NO. SHO 59[1984]-24134

Int. Cl.^{3}:	F 24 F 11/02 G 01 K 17/00
Sequence Nos. for Office Use:	L 7914-3L 7269-2F
Filing No.:	Sho 57[1982]-134257
Filing Date:	July 31, 1982
Publication Date:	February 7, 1984
No. of Inventions:	1 (Total of 2 pages)
Examination Request:	Not filed

CALORIMETER FOR AIR CONDITIONER

[Kuchokiyo karorimeta]

Inventor:	Yukata Momose
Applicant:	Aishin Seiki K.K.

[There are no amendments to this patent.]

Claim

A type of calorimeter for an air conditioner characterized by the fact that it has an intermediate heat exchanger for heat exchange between the warm water circuit of the evaporation-side heat exchanger and the cold water circuit of the condensation-side heat exchanger, and a cooling heat exchanger is set in said cold water circuit for heat exchange with the tap water.

Detailed explanation of the invention

The present invention pertains to a type of calorimeter for an air conditioner for measurement of the ability of the air conditioner/heater.

Figure 1 shows the calorimeter for air conditioner of the prior art for measurement of the ability of air conditioner/heater. In the air conditioner system that has compressor (1), condenser (2), expansion valve (3) and evaporator (4), there is cold water circuit (6) having cooling column (5) on the condensation heat-releasing side. Condenser (2) is used to perform heat exchange. Also, on the evaporation heat-absorbing side, there is warm water circuit (9) having warm water tank (8) with heater (7) contained in it. By means of evaporator (4), heat exchange is performed. The temperatures before and after heat exchange of condenser (2) of cold water circuit (6) and the temperatures before and after heat exchange in evaporator (4) of warm water circuit (3) are measured to perform calorie computation. For said conventional calorimeter, there should be cooling column (5) for forming cooling water and warm water tank (8) containing heater (7). Depending on the capacity of the air conditioner system, said cooling column and warm water tank may have a large size, and the energy consumption is also high. The high cost is undesired.

In order to solve the aforementioned problem of high cost, the present invention provides a technical means that exploits the heat released on the condensation side during heat absorption on the evaporation side by means of an intermediate heat exchanger for heat exchange between the heat released on the condensation side and the heat absorbed on the evaporation side, and a device that cools the difference between the released heat and the absorbed heat.

According to the present invention, the calorimeter can be made smaller in size and easier in operation. Also, as cooling is performed simply by means of the difference between the released heat and the absorbed heat, the cooling mechanism is also simpler.

In the following, an explanation will be given regarding an application example of the present invention with reference to Figure 2.

The creon [sic; Freon] gas from condenser (1) passes through condenser (2), expansion valve (3) and evaporator (4), and enters compressor (1). Cold water circuit (10) on the condensation heat releasing side has a heat exchanger portion in condenser (2), as well as intermediate heat exchanger (11) and cooling heat exchanger (12). Also, warm water circuit (13) on the evaporation heat absorbing side has heat exchanger portion in the evaporator and intermediate heat exchanger (11). Pump (14) and auxiliary tank (15) are set in cold and warm water circuits (10), (13), respectively. Also, bypass valve (16) is set in the portion of intermediate heat exchanger (11), and bypass filter valve (16) is set in the portion of cooling heat exchanger (12). (17) represents a flow-rate adjusting valve.

The Freon gas from compressor (1) is condensed in condenser (2), with heat released. The released heat is transferred with cold water circuit (10) to warm water circuit (13) by means of intermediate heat exchanger (11). The Freon gas passes through expansion value (3) and is evaporated in evaporator (4). Heat absorption in this case is provided with the heat released on the condensation side by means of intermediate heat exchanger (11). On the other hand, because

the released heat quantity is larger than the absorbed heat quantity by the power of compressor (1), the cooling water passing through intermediate heat exchanger (11) is cooled in cooling heat exchanger (12) to eliminate the heat quantity of such difference. For this cooling operation, as the heat quantity is small, tap water can be used to realize the purpose well, and there is no need to make use of a cooling column. The calorie may be determined by measuring the temperatures of the cooling water at the inlet and outlet of condenser (2) and cooling heat exchanger (12), and the temperatures of the warm water at the inlet and outlet of evaporator (4).

Brief description of the figures

Figure 1 is a diagram illustrating the calorimeter in the prior art. Figure 2 is a diagram illustrating the calorimeter in an example of the present invention.

Explanation of the symbols

1 Compressor
2 Condenser
3 Expansion valve
4 Evaporator
10 Cold water circuit
11 Intermediate heat exchanger
12 Cooling heat exchanger
13 Warm water circuit

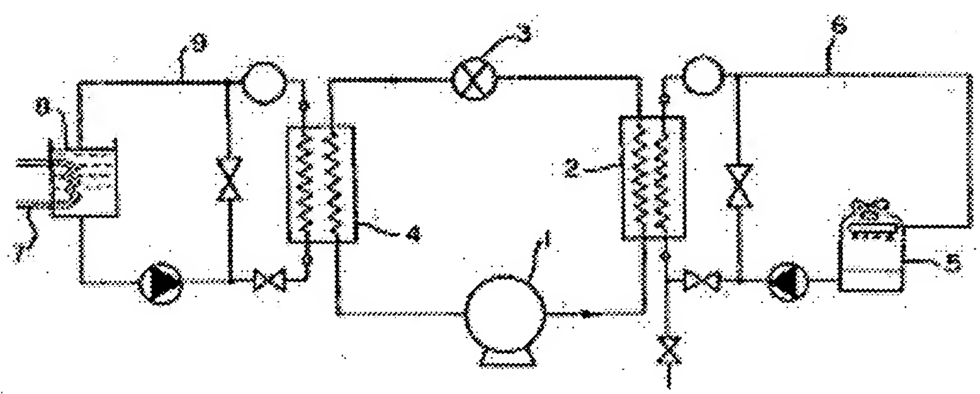


Figure 1

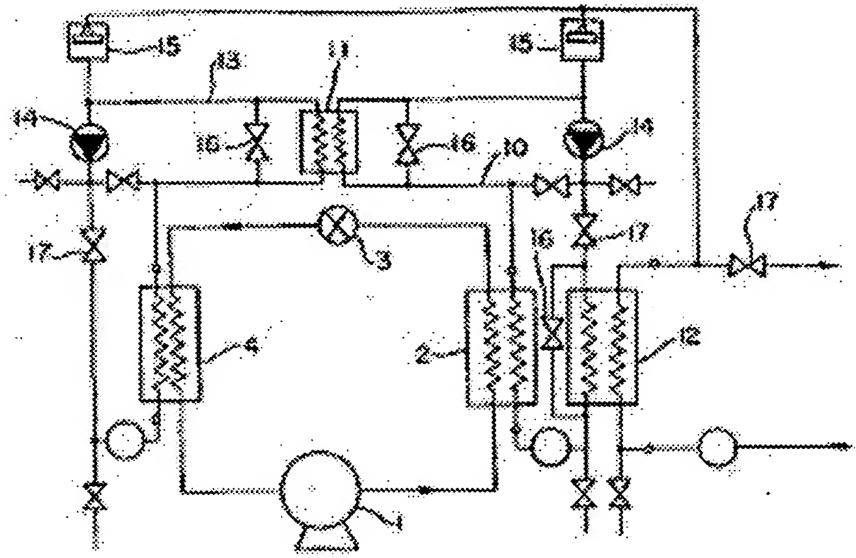


Figure 2

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(19)日本国特許庁（ＪＰ）　(12) 公開特許公報(A)　(11)特許出願公開番号

特開平10－76837

(43)公開日　平成10年(1998) 3月24日

(51)Int.Cl.⁶	識別記号	庁内整理番号	F I		技術表示箇所
B60H 1/03			B60H	1/03	Z
1/22				1/22	

審査請求　未請求　請求項の数3　OL　(全 6 頁)

(21)出願番号　特願平8－236638

(22)出願日　平成8年(1996) 9月6日

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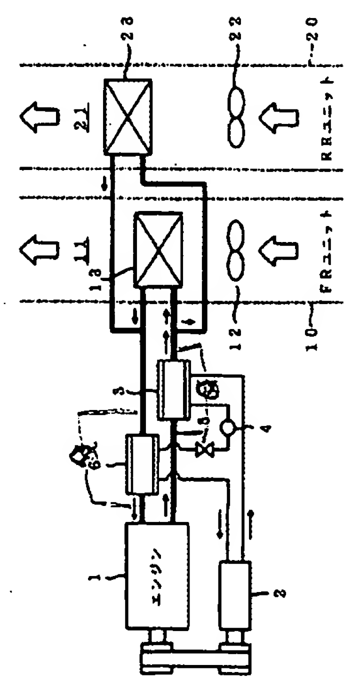
(54)【発明の名称】　自動車用暖房装置

(57)【要約】

【課題】フロントとリヤ側同時に暖房性能の向上が可能で、燃費が向上し、故障時のリカバリーが可能な「自動車用暖房装置」を提供する。

【解決手段】フロントユニット１０とリヤユニット２０内にヒータコア１３、２３をそれぞれ設け、各ヒータコア１３、２３の温水入口とエンジン１の温水出口との間に、高温高圧の冷媒を利用して各ヒータコア１３、２３へ流入する温水を加熱するコンデンサ３を設けるとともに、各ヒータコア１３、２３の温水出口とエンジン１の温水入口との間に、低温低圧の冷媒を利用して各ヒータコア１３、２３から流出した温水を冷却するエバポレータ６を設ける。



【特許請求の範囲】

【請求項1】取り入れた空気を車室内に向かって送る通風路（11、21）内に、エンジン（1）を冷却するための温水を利用して取り入れ空気を加熱するヒータコア（13、23）を設け、当該ヒータコア（13、23）の温水入口と前記エンジン（1）の温水出口との間に、冷凍サイクルを構成するコンプレッサ（2）から吐出された冷媒を利用して前記ヒータコア（13、23）へ流入する温水を加熱する第1熱交換器（3）を設け、前記ヒータコア（13、23）の温水出口と前記エンジン（1）の温水入口との間に、前記第1熱交換器（3）から流出した冷媒を利用して前記ヒータコア（13、23）から流出した温水を冷却する第2熱交換器（6）を設けたことを特徴とする自動車用暖房装置。

【請求項2】それぞれ取り入れた空気を車室内の前方領域および後方領域に向かって送る第1通風路（11）および第2通風路（21）を有し、前記ヒータコア（13、23）をそれぞれ前記第1通風路（11）および前記第2通風路（21）に配設したことを特徴とする請求項1記載の自動車用暖房装置。

【請求項3】前記第1熱交換器（3）および前記第2熱交換器（6）をリキッドタンク（4）および膨脹弁（5）と共に一体化したことを特徴とする請求項1または2記載の自動車用暖房装置。

【発明の詳細な説明】

【0001】

【発明の属する技術分野】本発明は、冷凍サイクル内の高温高圧の冷媒を利用して温水を加熱するようにした自動車用暖房装置に関する。

【0002】

【従来の技術】たとえば、最近の一部の高級車や比較的車室内空間が大きいワンボックスカーなどには、室内全体について快適な空調状態が得られるよう、車室内の前方領域（たとえば、前席部分）はフロントユニットにより、後方領域（たとえば、第2席、第3席等の後席部分）はリヤユニットによりそれぞれ独立に空気調和する、通常デュアルエアコンと称される自動車用空気調和装置が搭載されている。

【0003】この種の自動車用空気調和装置として、たとえば、暖房運転時において、フロントユニットはエンジン冷却水を熱源として利用するが、リヤユニットはコンプレッサにより圧縮された高温高圧の冷媒を熱源として利用するようにしたシステムがある。なお、この種の装置は、冷媒の循環過程（冷凍サイクル）において低温の外部空気から熱を汲み上げて車室内を暖房することから、ヒートポンプ式の自動車用空気調和装置と称されている。

【0004】ところが、この種の装置で暖房運転をする場合、たとえば、冬季の朝のように外気温度が低いときには、起動時にエンジン冷却水の温度が低く、また、冷媒の温度の上昇速度も俊敏でないため、運転開始と同時に暖かい空気が吹き出されるような状態になりにくく、いわゆる即暖性が不十分となり、また、暖房性能も不足気味となるおそれがある。特に、ディーゼルエンジンを搭載した車室内空間の大きいワンボックスカーでは、通常のガソリンエンジン車と比べてエンジン冷却水の温度上昇が遅く、しかも広い空間を暖房しなければならないことから、即暖性、暖房性能ともに不足する傾向がある。

【0005】そこで、現在では、エンジン冷却水の熱を利用して冷媒を加熱し、エンタルピーが増加したより高温の冷媒を用いて、より高い暖房性能を発揮するようにしたヒートポンプ式自動車用空気調和装置が開発されている（たとえば、特願平7－271621号参照）。

【0006】

【発明が解決しようとする課題】しかしながら、フロントユニットは温水を利用しリヤユニットは高温高圧の冷媒を利用してそれぞれ空気を加熱する上記のヒートポンプ式システムにあっては、次のような懸念事項がある。

【0007】第1に、特にリヤユニットについて、一般に冷凍サイクルにおいてはコンプレッサ保護のためコンプレッサ吐出圧力が上昇した時にコンプレッサのON／OFF制御を行って吐出圧力を下げるようにしているため、冷媒温度の上昇に一定の限界があり、したがって、それとの熱交換により加熱される空気の温度、つまり吹出風の温度にも一定の限界がある（たとえば、コンプレッサの信頼性を考慮すると最大で６０℃程度の吹出口温度しか得られない）。

【0008】第2に、暖房時（低外気温度時）のみならず温調時（中間外気温度時）においても常時コンプレッサを作動させるため、専ら温水を利用して空気を加熱する通常の暖房システムに対し燃費が低下するおそれがある。

【0009】第3に、もし万が一コンプレッサ等が故障した場合には、リヤユニットは一切機能しなくなり、後席の暖房ができなくなってしまう。

【0010】第4に、フロントユニットとリヤユニットとでそれぞれ別個の熱源（前者は温水、後者は高温高圧の冷媒）を利用するため、1つの手段でフロントユニットおよびリヤユニットについて同時に暖房性能の向上を図ることができない。

【0011】ところで、たとえば寒冷地などにおいては、自動車用空調装置として車室内を暖房する装置（自動車用暖房装置）のみを搭載した車両も多く、そうした自動車用暖房装置に対する需要も多いことから、専ら暖房にのみ着目し、上記の懸念事項を解決できる自動車用暖房装置の開発が強く求められていた。

【0012】本発明は、本出願人が現在開発中のヒートポンプ式自動車用空気調和装置における上記課題に着目してなされたものであり、既存の温水式ヒータを基本と

しつつ、上記の懸念事項をすべて解決しうる全く新しい自動車用暖房装置を提供することを目的とする。

【0013】

【課題を解決するための手段】上記目的を達成するため、請求項1記載の発明は、取り入れた空気を車室内に向かって送る通風路内に、エンジンを冷却するための温水を利用して取り入れ空気を加熱するヒータコアを設け、当該ヒータコアの温水入口と前記エンジンの温水出口との間に、冷凍サイクルを構成するコンプレッサから吐出された冷媒を利用して前記ヒータコアへ流入する温水を加熱する第1熱交換器を設け、前記ヒータコアの温水出口と前記エンジンの温水入口との間に、前記第1熱交換器から流出した冷媒を利用して前記ヒータコアから流出した温水を冷却する第2熱交換器を設けたことを特徴とする。

【0014】この発明にあっては、エンジンからヒータコアへ流れる温水は、第1熱交換器にて、コンプレッサから吐出された冷媒を利用して、つまり、コンプレッサから吐出された高温高圧の冷媒との熱交換（サイクルからの放熱）により加熱され、その温度が上昇する。これにより、ヒータコアにはより高温の温水が流れるようになるため、ヒータコアを通過する空気はより高温に加熱され、吹出風温度が上昇する。また、ヒータコアからエンジンへ戻る温水は、第2熱交換器にて、第1熱交換器から流出した冷媒を利用して、つまり、第1熱交換器から流出しリキッドタンクを経て膨脹弁で断熱膨脹された低温低圧の気化しやすい冷媒との熱交換（サイクルへの吸熱）により冷却された後、エンジンに帰還する。すなわち、いわゆるヒートポンプ式ヒータのように高温高圧の冷媒を利用して直接空気を加熱するのではなく間接的に温水を加熱するものではあるが、温水を加熱することによってヒータコアを流れる温水温度が上昇することから、通常の温水式ヒータと比べて暖房性能が向上する。また、100℃近くまで温度上昇しうる温水を吹出風の熱源とするため、コンプレッサ吐出圧力（Pd）制御により冷媒温度の上昇に一定の限界があるヒートポンプ式ヒータと比べても暖房性能が向上する。

【0015】また、コンプレッサを停止すれば通常の温水式ヒータとして作動するため、温水の温度が十分に上昇した後コンプレッサを停止するようにすれば、常時コンプレッサを作動させる必要はなくなる。

【0016】また、コンプレッサ等が故障した場合でも、冷媒との熱交換による温水の加熱はできなくなるものの、通常の温水式ヒータとしてはなお作動しうるため、依然として暖房自体は可能である。

【0017】請求項2記載の発明は、上記請求項1記載の発明において、それぞれ取り入れた空気を車室内の前方領域および後方領域に向かって送る第1通風路および第2通風路を有し、前記ヒータコアをそれぞれ前記第1通風路および前記第2通風路に配設したことを特徴とする。

【0018】この発明にあっては、車室内の前方領域（フロント側）および後方領域（リヤ側）共に冷媒との熱交換で加熱された温水によって暖房される。これにより、1つの手段でフロント側とリヤ側について同時に暖房性能の向上が図られる。

【0019】請求項3記載の発明は、上記請求項1または2記載の発明において、前記第1熱交換器および前記第2熱交換器をリキッドタンクおよび膨脹弁と共に一体化したことを特徴とする。

【0020】この発明にあっては、第1熱交換器、第2熱交換器、リキッドタンク、および膨脹弁が一体化され、スペース効率の有効化が図られる。たとえば、車両の床下スペースに1ユニットで取り付けることができる。

【0021】

【発明の実施の形態】以下、図面を使って、本発明の実施の形態を説明する。図1は本発明に係る自動車用暖房装置の一実施形態を示す概略構成図である。この自動車用暖房装置は、送風機により取り入れた内外気を加熱して車室内の前席および後席に向かってそれぞれ吹き出すフロントユニット10とリヤユニット20とを有している。

【0022】これら両ユニット10、20は、全く同じ構成を有しており、従来一般の自動車用暖房装置と同様、それぞれ、ケーシング内に形成された通風路11、21内に、白抜き矢印で示す空気の流れ方向の上流側から順に、送風機12、22およびヒータコア13、23を配設して構成されている。以下、フロントユニット10内のヒータコア13をフロントヒータコア、リヤユニット20内のヒータコア23をリヤヒータコアとそれぞれ呼ぶことにする。ヒータコア13、23は、エンジン1の冷却水（温水）を利用して取り入れ空気を加熱する熱交換器である。なお、図示しないが、より詳細には、両ユニット10、20は上流側から順にインテークユニットとヒータユニットを有し、両者をヒータダクトで連結して構成されている。インテークユニットにはインテークドアと前記送風機12、22が配置され、ヒータユニットにはエアミックスドアと前記ヒータコア13、23が配置されている。エアミックスドアは、ヒータコア13、23の前面に設けられ、ヒータコア13、23を通過した温風とこれを迂回した冷風（外気）との比率を調節してヒータコア13、23の下流域で所望温度の空気を作ったり、あるいはヒータコア13、23に空気が流通しないようにしている。また、ヒータユニットのヒータコア13、23下流側には、それぞれ、車室内の前席および後席への各種吹出口が形成されている。

【0023】エンジン1と各ユニット10、20内のヒータコア13、23とはそれぞれ温水配管で連結されており、エンジンから出た温水は途中で分流して各ヒータ

コア13、23へ流入し、各ヒータコア13、23から流出した温水は途中で合流してエンジン1へ戻るようになっている。また、図示しないが、各ヒータコア13、23の温水入口にはウォータコックがそれぞれ設けられており、このウォータコックを開状態にすることによってエンジン1から各ヒータコア13、23へそれぞれ温水が導入されるようになっている。

【0024】また、両ユニット10、20の外部には、エンジン1によって回転駆動されるコンプレッサ2と、第1熱交換器として機能する温水循環可能なコンデンサ3と、リキッドタンク4と、膨脹弁5と、第2熱交換器として機能する温水循環可能なエバポレータ6とが配設されている。冷凍サイクルは、これらコンプレッサ2、コンデンサ3、リキッドタンク4、膨脹弁5、およびエバポレータ6を配管で連結して構成されている。たとえば、コンデンサ3およびエバポレータ6は、概略、図2に示すような構造をしており、おのおのの内部には温水が流れる通路と冷媒が流れる通路とがそれぞれ形成され、温水と冷媒との間で熱交換が行われるようになっている。

【0025】コンデンサ3は、ヒータコア13、23の温水入口(の分流点)とエンジン1の温水出口との間に設けられており、エンジン1から各ヒータコア13、23へ流れる温水を、コンプレッサ2から吐出された冷媒を利用して、つまり、コンプレッサ1から吐出された高温高圧の冷媒との熱交換(サイクルからの放熱)によって加熱する機能を有している。

【0026】一方、エバポレータ6は、ヒータコア13、23の温水出口(の合流点)とエンジン1の温水入口との間に設けられており、各ヒータコア13、23からエンジン1へ戻る温水を、コンデンサ3から流出した冷媒を利用して、つまり、コンデンサ3から流出しリキッドタンク4を経て膨脹弁5で断熱膨脹された低温低圧の気化しやすい冷媒との熱交換(サイクルへの吸熱)によって冷却する機能を有している。

【0027】次に、作用を説明する。暖房時、フロントユニット10およびリヤユニット20共に、各ウォータコック(図示せず)を開にすると、エンジン1からコンデンサ3を通ってフロントヒータコア13およびリヤヒータコア23へそれぞれ温水が流入し、その後、内部を流通した後、フロントヒータコア13およびリヤヒータコア23からそれぞれ流出しエバポレータ6を通ってエンジン1に帰還する。このとき、コンプレッサ2をONすると、コンプレッサ2→コンデンサ3→リキッドタンク4→膨脹弁5→エバポレータ6→コンプレッサ2と冷媒が循環する冷凍サイクルが成立し、コンデンサ3およびエバポレータ6がそれぞれ機能する。すなわち、過熱蒸気の状態でコンプレッサ2に吸入された冷媒はコンプレッサ2により断熱圧縮されて高温高圧のガス冷媒となってコンプレッサ2から吐出される。この高温高圧のガス冷媒はコンデンサ3に導かれ、ここでエンジン1から流入した温水に熱を放出してその温水を加熱し、中温高圧の液冷媒となる。ついで、この液冷媒はリキッドタンク4を経て、膨脹弁5を通過することにより断熱膨脹して気化しやすい気液混合冷媒となった後、エバポレータ6に導かれ、ここで各ヒータコア13、23から流出した温水から熱を吸収してその温水を冷却し、蒸発しつつ等圧膨脹を続け、過熱蒸気となって再びコンプレッサ2に吸入される。したがって、エンジン1から各ヒータコア13、23へ流れる温水は、コンデンサ3によって高温高圧のガス冷媒との熱交換により加熱されてその温度が上昇するため、各ヒータコア13、23にはより高温の温水が流れるようになり、各ヒータコア13、23の暖房性能が向上する。その結果、各ヒータコア13、23を通過する空気はより高温に加熱されることになる。また、各ヒータコア13、23からエンジン1へ戻る温水は、エバポレータ6によって低温低圧の気化しやすい冷媒との熱交換により冷却されてその温度が低下するため、エンジン1の冷却効果が高まる。なお、コンプレッサ2をOFFすると、上記の冷凍サイクルは成立せず、温水を利用した通常の温水式ヒータとして作動する。

【0028】このとき、各送風機12、22によりフロントユニット10およびリヤユニット20内にそれぞれ取り込まれた空気は、各ヒータコア13、23において加熱された後、流下し、所定の吹出口から車室内の前席および後席に吹き出される。その際、前席と後席とは各エアミックスドア(図示せず)の開度を調節することによってそれぞれ独立に暖房または温調される。

【0029】また、温水の温度が十分に上昇した後は、不必要に温水を加熱しないよう、コンプレッサ2をOFFしてヒートポンプシステムを停止させ、通常の温水式ヒータとして作動させる。

【0030】なお、暖房初期においては、吹出風温度がある程度高くなるまで、各エアミックスドア(図示せず)により空気が各ヒータコア13、23を通過しないようにする制御を加えることも可能である。

【0031】また、温調時には、各エアミックスドア(図示せず)の開度を調節して各ヒータコア13、23で加熱された温風とそれを迂回した冷風(外気)とのミックス割合を調節して吹出口温度の調整を行う。

【0032】また、暖房を行う必要がない場合(たとえば、夏場)には、各ウォータコック(図示せず)を閉にして、各ヒータコア13、23に温水が導入されないようにしておく。

【0033】したがって、本実施形態によれば、いわゆるヒートポンプ式ヒータのように高温高圧の冷媒を利用して直接空気を加熱するのではなく間接的に温水を加熱するものではあるが、冷媒により温水を加熱することによってヒータコア13、23を流れる温水温度が上昇するため、通常の温水式ヒータと比べて暖房性能が向上す

る。また、１００℃近くまで温度上昇しうる温水を吹出風の熱源とするため、上記したようにコンプレッサ吐出圧力（Pd）制御により冷媒温度の上昇に一定の限界があるヒートポンプ式ヒータと比べても、暖房性能が向上する。たとえば、実験によれば、最大で７０℃以上の吹出口温度が得られた。

【００３４】また、コンプレッサ２をＯＦＦすれば通常の温水式ヒータとして作動するため、温水温度が十分に上昇した後コンプレッサ２を停止するようにすれば、常時コンプレッサ２を作動させる必要がなくなり、燃費の向上を図ることができる。

【００３５】さらに、コンプレッサ２等が故障した場合でも、冷媒との熱交換による温水の加熱はできなくなるものの、通常の温水式ヒータとしてはなお作動しうるため、依然として暖房自体は可能である。つまり、コンプレッサ２およびシステム故障時のリカバリーが可能である。

【００３６】また、フロントユニット１０およびリヤユニット２０共に冷媒との熱交換により加熱された温水によって暖房可能であるため、フロント側とリヤ側について同時に暖房性能の向上を図ることができる。

【００３７】なお、スペース効率の有効化とコストの低減を図るためには、コンデンサ３、エバポレータ６、リキッドタンク４、および膨張弁５を一体化することが好ましい。図3はその一体化したユニットの一例を示す外観図、図4は同ユニットの構造を示す模式図である。ここで、図４（Ａ）は温水の流れる構造を示し、図４（Ｂ）は冷媒の流れる構造を示している。

【００３８】図中、３０は一体化されたユニット、３１はコンデンサ３に相当するコンデンサ部、３２はリキッドタンク４に相当するリキッドタンク部、３３はエバポレータ６に相当するエバポレータ部である。膨張弁５はユニット３０のエバポレータ部３３に取り付けられている。

【００３９】このように、コンデンサ３、エバポレータ６、リキッドタンク４、および膨張弁５を一体化した場合には、車両の床下スペースに１ユニットで取り付けることができ、省スペース化が可能で、低コスト化も図られる。また、リキッドタンク４とエバポレータ６を一体化してリキッドタンク部３２とエバポレータ部３３を接触させたため、両者の間で熱交換が行われ、性能が向上する。

【００４０】

【発明の効果】以上述べたように、請求項１記載の発明によれば、高温高圧の冷媒を利用して温水を加熱するので、ヒータコアを流れる温水温度が上昇し、暖房性能が向上する。また、コンプレッサを停止しても通常の温水式ヒータとして作動するため、常時コンプレッサを作動させる必要がなく、燃費の向上が図られる。さらに、コンプレッサ等が故障した場合でも、通常の温水式ヒータとしてはなお作動可能であるため、依然として暖房自体は可能である。

【００４１】請求項２記載の発明によれば、上記請求項１記載の発明の効果に加え、車室内のフロント側およびリヤ側共に冷媒との熱交換により加熱された温水によって暖房可能としたので、１つの手段でフロント側とリヤ側について同時に暖房性能の向上を図ることができる。

【００４２】請求項３記載の発明によれば、上記請求項１または２記載の発明の効果に加え、第１熱交換器、第２熱交換器、リキッドタンク、および膨張弁を一体化したので、スペース効率の有効化およびコストの低減が図られる。

【図面の簡単な説明】

【図１】本発明に係る自動車用暖房装置の一実施形態を示す概略構成図である。

【図２】図１に示すコンデンサおよびエバポレータの構造を示す外観図である。

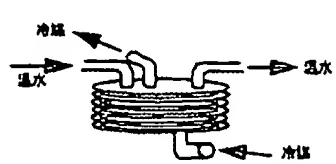
【図３】一体化されたユニットの一例を示す外観図である。

【図４】同ユニットの構造を示す模式図である。

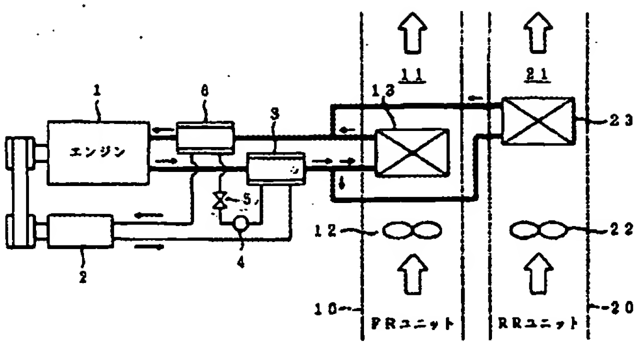
【符号の説明】

１…エンジン
２…コンプレッサ
３…コンデンサ（第１熱交換器）
４…リキッドタンク
５…膨張弁
６…エバポレータ（第２熱交換器）
１１、２１…通風路
１３、２３…ヒータコア
３０…一体化されたユニット

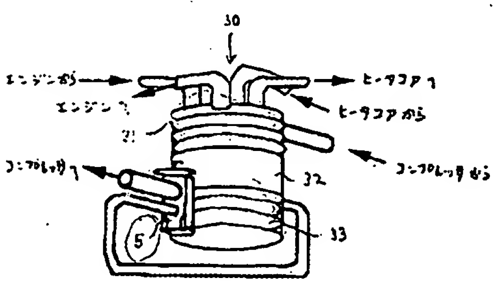
【図2】



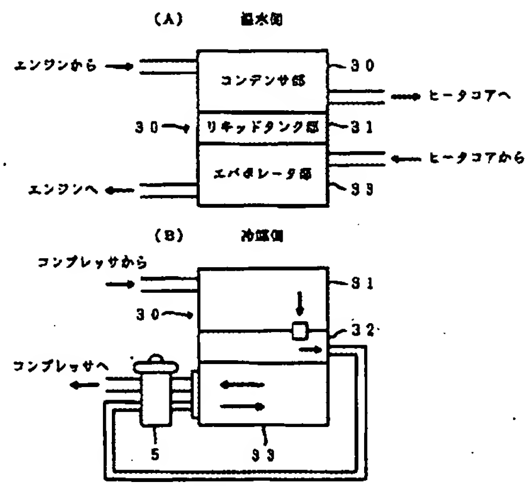
【図1】



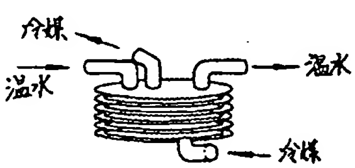
【図3】



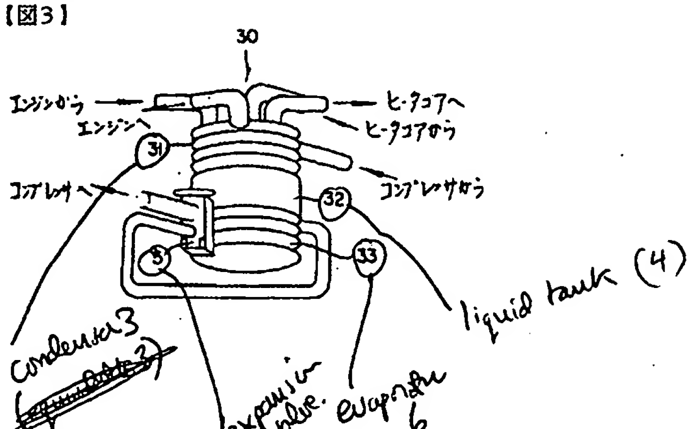
【図4】



【手続補正書】
【提出日】平成8年12月9日
【手続補正1】
【補正対象書類名】図面
【補正対象項目名】図2
【補正方法】変更
【補正内容】
【図2】



【手続補正2】
【補正対象書類名】図面
【補正対象項目名】図3
【補正方法】変更
【補正内容】
【図3】



CLIPPEDIMAGE= JP410076837A
PAT-NO: JP410076837A
DOCUMENT-IDENTIFIER: JP 10076837 A
TITLE: HEATING SYSTEM FOR AUTOMOBILE

PUBN-DATE: March 24, 1998

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APPL-NO: JP08236638
APPL-DATE: September 6, 1996

INT-CL_(IPC): B60H001/03; B60H001/22

ABSTRACT:
PROBLEM TO BE SOLVED: To provide an automobile heating system that is capable of improvement in heating performance simultaneously at both front and rear sides, improvement in fuel consumption and recovery at time of trouble.

SOLUTION: Each of heater cores 13 and 23 is installed in both front and rear units 10 and 20, and a condenser 3 heating hot water to flow into these heater cores 13 and 23 in use of a high temperature-pressure coolant is installed in space between each hot-water inlet of these cores 13, 23 and a hot-water outlet of an engine 1, while an evaporator 6 cooling the hot water run out of these heater cores 13 and 23 in use of a low temperature-pressure coolant is installed in space between each hot-water outlet of these cores 13, 23 and hot-water inlet of the engine 1 as well.

JP 10-76837

* NOTICES *

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Apl 8-236638

1.This document has been translated by computer. So the translation may not reflect the original precisely.
2.**** shows the word which can not be translated.
3.In the drawings, any words are not translated.

CLAIMS

[Claim(s)]
[Claim 1] In the ventilation flue (11, 21) to which the taken-in air is sent toward the vehicle interior of a room The heater core (13, 23) which takes in using the warm water for cooling an engine (1), and heats air is prepared. between the warm water inlet of the concerned heater core (13, 23), and the warm water outlet of the aforementioned engine (1) The 1st heat exchanger (3) which heats the warm water which flows into the aforementioned heater core (13, 23) using the refrigerant breathed out from the compressor (2) which constitutes a refrigerating cycle is formed. between the warm water outlet of the aforementioned heater core (13, 23), and the warm water inlets of the aforementioned engine (1) -- the outflow from the 1st aforementioned heat exchanger (3) -- the bottom -- a refrigerant -- using -- the outflow from the aforementioned heater core (13, 23) -- the heating apparatus for automobiles characterized by forming the 2nd heat exchanger (6) which cools warm water the bottom
[Claim 2] The heating apparatus for automobiles according to claim 1 characterized by having the 1st ventilation flue (11) and the 2nd ventilation flue (21) to which the air taken in, respectively is sent toward the front field and back field of the vehicle interior of a room, and arranging the aforementioned heater core (13, 23) in the 1st aforementioned ventilation flue (11) and the 2nd aforementioned ventilation flue (21), respectively.
[Claim 3] The heating apparatus for automobiles according to claim 1 or 2 characterized by unifying the 1st aforementioned heat exchanger (3) and the 2nd aforementioned heat exchanger (6) with a liquid tank (4) and an expansion valve (5).

[Translation done.]

DETAILED DESCRIPTION

[Detailed Description of the Invention]
[0001]
[The technical field to which invention belongs] this invention relates to the heating apparatus for automobiles which heated warm water using the refrigerant of the elevated-temperature hyperbaric pressure within a refrigerating cycle.
[0002]
[Description of the Prior Art] For example, the conditioner for automobiles which carries out the front field (for example, front seat fraction) of the vehicle interior of a room by the front unit, and carries out air conditioning of the back field (for example, backseat fractions, such as the 2nd seat and the 3rd seat) independently by the rear unit, respectively and which is usually called a dual air-conditioner is carried so that the air-conditioning status comfortable about the whole interior of a room may be acquired by a part of latest luxury cars and one-box cars with comparatively large vehicle indoor space.
[0003] Although a front unit uses an engine cooling water as a heat source as this kind of a conditioner for automobiles for example, at the time of heating operation, a rear unit has the system which used the refrigerant of the elevated-temperature hyperbaric pressure compressed by the compressor as a heat source. In addition, since this kind of equipment pumps up heat from low-temperature exterior air in the cyclic process (refrigerating cycle) of a refrigerant and heats the vehicle interior of a room, it is called the conditioner for automobiles of a heat pump formula.
[0004] However, like the morning of winter, when carrying out heating operation with this kind of equipment, when an OAT is low, there is a possibility that it may be hard to be in the status that warm air blows off simultaneously with a start up at the time of activation since the temperature of an engine cooling water is low and the climbing speed of the temperature of a refrigerant does not have it, either, and the so-called ****** may become inadequate, and a heating performance may also serve as insufficient feeling. [quick] Especially in the large one-box car of vehicle indoor space which carried the diesel power plant, since the temperature rise of an engine cooling water must heat large space late moreover compared with a usual gasoline-engine vehicle, there is an inclination which ****** and a heating performance run short of.
[0005] Then, at present, a refrigerant is heated using the heat of an engine cooling water, and the conditioner for heat pump formula automobiles which enthalpy increases and demonstrated the higher heating performance using the refrigerant of a reliance elevated temperature is developed (for example, refer to Japanese Patent Application No. 271621 [seven to]).
[0006]
[Problem(s) to be Solved by the Invention] However, a front unit uses warm water, and if a rear unit is in the above-mentioned heat pump formula system which heats air using the refrigerant of the elevated-temperature hyperbaric pressure, respectively, it has the following concern matters.
[0007] When a compressor discharge pressure generally goes up to the 1st in a refrigerating cycle especially about a rear unit for compressor protection, in order to perform ON/OFF control of a compressor to it and to lower a discharge pressure to it, A fixed limitation is in the temperature of the blow-off style of the air which a fixed limitation is in elevation of a coolant temperature, therefore is heated by the heat exchange with that, i.e., temperature, (for example, consideration of the reliability of a compressor obtains only the outlet temperature of about 60 degrees C by the maximum).
[0008] There is a possibility that mpg may fall to it to the usual heating system which heats air chiefly using warm water in order to always operate a compressor to the 2nd not only at the time (at the time of a low OAT) of heating but at the time (at the time of an interval OAT) of a ** tone.
[0009] When a compressor etc. should break down to the 3rd, a rear unit stops functioning at all and the heating of a backseat of it will become impossible.
[0010] Since a respectively separate heat source (refrigerant of the elevated-temperature hyperbaric pressure [former / latter / warm water and]) is used for the 4th in a front unit and a rear unit, enhancement in a heating performance cannot be simultaneously aimed at about a front unit and a rear unit with one means.
[0011] Only paying attention to heating, it was chiefly asked for the development of the heating apparatus for automobiles which can solve the above-mentioned concern matter strongly from there being also many vehicles only carrying the equipment (heating apparatus for automobiles) which heats the vehicle interior of a room by the place as an air conditioner for automobiles in a cold district etc., and there being much need over such a heating apparatus for automobiles.
[0012] These people are made now paying attention to the above-mentioned technical probrem in the conditioner for heat pump formula automobiles under development, and on the basis of the existing warm water formula heater, this invention aims at

offering the completely new heating apparatus for automobiles which can solve all the above-mentioned concern matters, while.
[0013]
[Means for Solving the Problem] In order to attain the above-mentioned purpose, invention according to claim 1 In the ventilation flue to which the taken-in air is sent toward the vehicle interior of a room, the heater core which takes in using the warm water for cooling an engine, and heats air is prepared. between the warm water inlet of the concerned heater core, and the warm water outlet of the aforementioned engine The 1st heat exchanger which heats the warm water which flows into the aforementioned heater core using the refrigerant breathed out from the compressor which constitutes a refrigerating cycle is formed. between the warm water outlet of the aforementioned heater core, and the warm water inlet of the aforementioned engine It is characterized by forming the 2nd heat exchanger which cools the warm water which flowed out of the aforementioned heater core using the refrigerant which flowed out of the 1st aforementioned heat exchanger.
[0014] If it is in this invention, the warm water which flows from an engine to a heater core is heated by the heat exchange (thermolysis from a cycle) with the refrigerant of the elevated-temperature hyperbaric pressure breathed out from the compressor with the 1st heat exchanger, using [that is,] the refrigerant breathed out from the compressor, and the temperature rises. Thereby, in order for hotter warm water to flow to a heater core, the air which passes a heater core is more heated by the elevated temperature, and blow-off wind temperature rises. Moreover, the warm water which returns from a heater core to an engine returns to an engine, after being cooled by the heat exchange (endothermic to a cycle) with the refrigerant which the low-temperature low voltage by which flows out of the 1st heat exchanger and heat insulation expansion was carried out with the expansion valve through the liquid tank tends to evaporate, using [that is,] the refrigerant which flowed out of the 1st heat exchanger with the 2nd heat exchanger. That is, although direct air is not heated like the so-called heat pump formula heater using the refrigerant of the elevated-temperature hyperbaric pressure but warm water is heated indirectly, compared with a usual warm water formula heater, a heating performance improves by heating warm water from the warm water temperature which flows a heater core rising. Moreover, in order to make into the heat source of the blow-off style the warm water which can carry out a temperature rise to about 100 degrees C, even if it compares with the heat pump formula heater which has a fixed limitation in elevation of a coolant temperature by compressor discharge-pressure (Pd) control, a heating performance improves.
[0015] Moreover, if it is made to suspend a compressor after the temperature of warm water fully rises in order to operate as a usual warm water formula heater, if a compressor is suspended, it is not necessary to make a compressor always come to operate.
[0016] Moreover, since it can operate in addition as a usual warm water formula heater although heating of the warm water by the heat exchange with a refrigerant becomes impossible even when a compressor etc. breaks down, the heating itself is still possible.

[0017] Invention according to claim 2 has the 1st ventilation flue and the 2nd ventilation flue to which the air taken in, respectively is sent in invention of the claim 1 above-mentioned publication toward the front field and back field of the vehicle interior of a room, and is characterized by arranging the aforementioned heater core in the 1st aforementioned ventilation flue and the 2nd aforementioned ventilation flue, respectively.
[0018] if it is in this invention -- the front field (front side) of the vehicle interior of a room, and a back field (rear **) -- it is heated by the warm water heated by both the heat exchanges with a refrigerant Thereby, enhancement in a heating performance is simultaneously achieved about rear ** a front side with one means.
[0019] Invention according to claim 3 is characterized by unifying the 1st aforementioned heat exchanger and the 2nd aforementioned heat exchanger with a liquid tank and an expansion valve in invention the above-mentioned claim 1 or given in two.
[0020] If it is in this invention, the 1st heat exchanger, the 2nd heat exchanger, a liquid tank, and an expansion valve are unified, and validation of space efficiency is attained. For example, it can attach in the under floor space of a vehicle in one unit.
[0021]
[Embodiments of the Invention] Hereafter, the gestalt of operation of this invention is explained using a drawing. Drawing 1 is an outline block diagram showing the 1 operation gestalt of the heating apparatus for automobiles concerning this invention. This heating apparatus for automobiles has the front unit 10 and the rear unit 20 which heat the inside-and-outside mind taken in with the blower, and blow off toward the front seat and backseat of the vehicle interior of a room, respectively.
[0022] Both [these] the units 10 and 20 have the completely same configuration, arrange the blowers 12 and 22 and the heater cores 13 and 23 in the ventilation flue 11 formed in casing, and 21 like the heating apparatus for automobiles of the general former sequentially from the upstream side of the direction of flow of the air shown by the white arrow head, respectively, and are constituted. Hereafter, the heater core 13 in the front unit 10 is made to call a front heater core and the heater core 23 in the rear unit 20 rear heater core, respectively. The heater cores 13 and 23 are heat exchangers which take in using the cooling water (warm water) of an engine 1, and heat air. In addition, although not illustrated, more, both the units 10 and 20 have an intake unit and a heater unit sequentially from an upstream side, connect both by the heater duct and are constituted by the detail. An intake door and the aforementioned blowers 12 and 22 are arranged at an intake unit, and an air mix door and the aforementioned heater cores 13 and 23 are arranged at the heater unit. An air mix door is prepared in the front face of the heater cores 13 and 23, the proportion with the cold blast (open air) which bypassed the warm air which passed the heater cores 13 and 23, and this is adjusted, and air is made not to circulate in the down-stream region of the heater cores 13 and 23 to making the air of request temperature ****, or the heater cores 13 and 23. Moreover, the various outlets to the front seat and backseat of the vehicle interior of a room are formed in heater core [of a heater unit] 13, and 23 lower-stream-of-a-river side, respectively.
[0023] the warm water which an engine 1, and each unit 10 and the heater cores 13 and 23 in 20 are connected for warm water

piping, respectively, and came out of the engine -- on the way -- coming out -- shunting -- each heater cores 13 and 23 -- flowing -- the outflow from each heater cores 13 and 23 -- the bottom -- warm water -- on the way -- it comes out, and it joins and returns to an engine 1 Moreover, although not illustrated, the water cock is prepared in the warm water inlet of each heater cores 13 and 23, respectively, and warm water introduces from an engine 1 to each heater cores 13 and 23 by changing this water cock into the open status, respectively.

[0024] Moreover, the compressor 2 by which a rotation drive is carried out with an engine 1, the capacitor 3 which functions as the 1st heat exchanger and in which hot water circulating is possible, the liquid tank 4, the expansion valve 5, and the evaporator 6 that functions as the 2nd heat exchanger and in which hot water circulating is possible are arranged in the exterior of both the units 10 and 20. A refrigerating cycle connects these compressors 2, the capacitor 3, the liquid tank 4, the expansion valve 5, and the evaporator 6 for piping, and is constituted. For example, an outline and structure which is shown in drawing 2 are carried out, the path where warm water flows, and the path where a refrigerant flows are formed in each interior, respectively, and a heat exchange performs the capacitor 3 and the evaporator 6 between warm water and a refrigerant.

[0025] The capacitor 3 is formed between the warm water inlet (******) of the heater cores 13 and 23, and the warm water outlet of an engine 1, and has the function to heat the warm water which flows from an engine 1 to each heater cores 13 and 23 by the heat exchange (thermolysis from a cycle) with the refrigerant of the elevated-temperature hyperbaric pressure breathed out from the compressor 1, using [that is,] the refrigerant breathed out from the compressor 2.

[0026] A refrigerant is used the bottom. the warm water which the evaporator 6 is formed between the warm water outlet (juncture) of the heater cores 13 and 23, and the warm water inlet of an engine 1, and returns from each heater cores 13 and 23 to an engine 1 on the other hand -- the outflow from a capacitor 3 -- That is, it has the function cooled by the heat exchange (endothermic to a cycle) with the refrigerant which the low-temperature low voltage by which flows out of a capacitor 3 and heat insulation expansion was carried out with the expansion valve 5 through the liquid tank 4 tends to evaporate.

[0027] Next, an operation is explained. If the front unit 10 and the rear unit 20 make each water cock (not shown) open at the time of heating, after warm water's flowing into the front heater core 13 and the rear heater core 23 through a capacitor 3 from an engine 1, respectively and circulating the interior after that, it flows out of the front heater core 13 and the rear heater core 23, respectively, and returns to an engine 1 through an evaporator 6. If a compressor 2 is turned on at this time, the refrigerating cycle through which the compressor 2 -> capacitor 3 -> liquid tank 4 -> expansion valve 5 -> evaporator 6 -> compressor 2 and a refrigerant circulate will be materialized, and the capacitor 3 and the evaporator 6 will function, respectively. That is, adiabatic compression of the refrigerant inhaled by the compressor 2 in the state of superheated steam is carried out by the compressor 2, it turns into the gas refrigerant of the elevated-temperature hyperbaric pressure, and is breathed out from a compressor 2. The gas refrigerant of this elevated-temperature hyperbaric pressure is led to a capacitor 3, emits heat to the warm water which flowed from the engine 1 here, heats the warm water, and turns into liquid cooling intermediation of the moderate temperature hyperbaric pressure. Subsequently, this liquid cooling intermediation passes through the liquid tank 4, after it turns into the vapor-liquid mixture refrigerant which carries out heat insulation expansion and is easy to evaporate by passing the expansion valve 5, absorbs heat from the warm water which is led to an evaporator 6 and flowed out of each heater cores 13 and 23 here, cools the warm water, it continues isotonic expansion, evaporating, serves as superheated steam, and is again inhaled by the compressor 2. Therefore, since it is heated by the heat exchange with the gas refrigerant of the elevated-temperature hyperbaric pressure and the temperature rises by the capacitor 3, hotter warm water comes to flow to each heater cores 13 and 23, and the heating performance of warm water which flows from an engine 1 to each heater cores 13 and 23 of each heater cores 13 and 23 improves. Consequently, the air which passes each heater cores 13 and 23 will be more heated by the elevated temperature. Moreover, since it is cooled by the heat exchange with the refrigerant to which low-temperature low voltage tends to evaporate the warm water which returns from each heater cores 13 and 23 to an engine 1 by the evaporator 6 and the temperature falls, the cooling effect of an engine 1 increases. In addition, if a compressor 2 is turned off, the above-mentioned refrigerating cycle will not be materialized but will operate as a usual warm water formula heater using warm water.

[0028] After heating the air incorporated in the front unit 10 and the rear unit 20 by each blowers 12 and 22, respectively at this time in each heater cores 13 and 23, it flows down and blows off from a predetermined outlet to the front seat and backseat of the vehicle interior of a room. Heating or the ** tone of a front seat and the backseat is independently carried out by adjusting the opening of each air mix door (not shown), respectively in that case.

[0029] Moreover, after the temperature of warm water fully rises, a compressor 2 is turned off, a heat pump system is stopped, and it is made to operate as a usual warm water formula heater so that warm water may not be heated superfluously.

[0030] In addition, in the early stages of heating, it is also possible to add the control whose air is made not to pass each heater cores 13 and 23 by each air mix door (not shown) until blow-off wind temperature becomes to some extent high.

[0031] Moreover, at the time of a ** tone, the mix rate with the cold blast (open air) which bypassed the warm air which adjusts the opening of each air mix door (not shown), and was heated with each heater cores 13 and 23, and it is adjusted, and outlet temperature is adjusted.

[0032] Moreover, when it is not necessary to heat (for example, summer), each water cock (not shown) is made close and warm water is made not to be introduced into each heater cores 13 and 23.

[0033] Therefore, although according to this operation gestalt direct air is not heated like the so-called heat pump formula heater using the refrigerant of the elevated-temperature hyperbaric pressure but warm water is heated indirectly, since the warm water temperature which flows the heater cores 13 and 23 by heating warm water by the refrigerant rises, a heating performance improves compared with a usual warm water formula heater. Moreover, even if it compares with the heat pump formula heater

which has a fixed limitation in elevation of a coolant temperature by compressor discharge-pressure (Pd) control as described above in order to make into the heat source of the blow-off style the warm water which can carry out a temperature rise to about 100 degrees C, a heating performance improves. For example, according to the experiment, the outlet temperature of 70 degrees C or more was obtained by the maximum.

[0034] Moreover, if it is made to suspend a compressor 2 after warm water temperature fully rises in order to operate as a usual warm water formula heater, if a compressor 2 is turned off, it is not necessary to make a compressor 2 always become unable to operate, and enhancement in mpg can be aimed at.

[0035] Furthermore, since it can operate in addition as a usual warm water formula heater although heating of the warm water by the heat exchange with a refrigerant becomes impossible even when a compressor 2 etc. breaks down, the heating itself is still possible. That is, the recovery at the time of the compressor 2 and a system failure is possible.

[0036] Moreover, since the warm water with which the front unit 10 and the rear unit 20 were heated by the heat exchange with a refrigerant can heat, enhancement in a heating performance can be simultaneously aimed at about rear ** a front side.

[0037] In addition, in order to aim at validation of space efficiency, and a reduction of a cost, it is desirable to unify a capacitor 3, the evaporator 6, the liquid tank 4, and the expansion valve 5. The external view showing [3] an example of the unified unit and the drawing 4 are ** type views showing the structure of this unit. Here, drawing 4 (A) shows the structure where warm water flows, and drawing 4 (B) shows the structure where a refrigerant flows.

[0038] The unit with which 30 were united, the capacitor section in which 31 is equivalent to a capacitor 3, the liquid tank section in which 32 is equivalent to the liquid tank 4, and 33 are the evaporator sections equivalent to an evaporator 6 among drawing. The expansion valve 5 is attached in the evaporator section 33 of a unit 30.

[0039] Thus, when a capacitor 3, the evaporator 6, the liquid tank 4, and the expansion valve 5 are unified, it can attach in the under floor space of a vehicle in one unit, and the formation of ** space is possible and low-cost-ization is also attained. Moreover, since the liquid tank 4 and the evaporator 6 were unified and the liquid tank section 32 and the evaporator section 33 were contacted, a heat exchange is performed among both and a performance improves.

[0040]

[Effect of the Invention] The warm water temperature which was described above and which flows a heater core since warm water is heated like using the refrigerant of the elevated-temperature hyperbaric pressure according to invention according to claim 1 rises, and a heating performance improves. Moreover, even if it suspends a compressor, in order to operate as a usual warm water formula heater, it is not necessary to always operate a compressor and enhancement in mpg is achieved. Furthermore, since it can still operate as a usual warm water formula heater even when a compressor etc. breaks down, the heating itself is still possible.

[0041] Since heating was made [according to invention according to claim 2] possible with the warm water with which front [of the vehicle interior of a room] and rear ** was heated by the heat exchange with a refrigerant in addition to the effect of the invention of the claim 1 above-mentioned publication, enhancement in a heating performance can be simultaneously aimed at about rear ** a front side with one means.

[0042] According to invention according to claim 3, since the 1st heat exchanger, the 2nd heat exchanger, the liquid tank, and the expansion valve were unified in addition to the effect of the invention the above-mentioned claim 1 or given in two, validation of space efficiency and a reduction of a cost are achieved.

[Translation done.]

PTO 05-4657 Patent No. Hei 10[1998]-76837

HEATING SYSTEM FOR AUTOMOBILE

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UNITED STATES PATENT AND TRADEMARK OFFICE
WASHINGTON, D.C. JULY 2005
TRANSLATED BY THE MCELROY TRANSLATION COMPANY

JAPANESE PATENT OFFICE
PATENT JOURNAL (BX)
KOKAI PATENT APPLICATION NO. HEI 10[1998]-76837

Int. Cl.[6]:	B 60 H 1/03 1/22
Filing No.:	Hei 8[1996]-236638
Filing Date:	September 6, 1996
Publication Date:	March 24, 1998
No. of Claims:	3 OL (Total of 6 pages)
Examination Request:	Not filed

HEATING SYSTEM FOR AUTOMOBILE

[Jidoshayo danbo sochi]

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[Amendments have been incorporated into the text of the translation.]

Claims

1. A heating system for automobile characterized by the fact that heater cores (13) and (23), which heat the intake air by using the hot water for cooling engine (1) were provided within ventilation trunks (11) and (21), which feed the intake air towards the car chamber, first heat exchanger (3), which heats the hot water flowing into aforementioned heater cores (13) and (23) by utilizing the coolant discharged from compressor (2) composing a refrigerating cycle was

provided between the hot water inlets of heater cores (13) and (23) and the hot water outlet of aforementioned engine (1), and second heat exchanger (6), which cools the hot water flowing out from aforementioned heater cores (13) and (23) by utilizing the coolant flowing out from aforementioned first heat exchanger (3) was provided between the hot water outlets of aforementioned heater cores (13) and (23) and the hot water inlet of aforementioned engine (1).

2. The heating system for automobile described in Claim 1 characterized by the fact that there is first ventilation trunk (11) and second ventilation trunk (21), which respectively feeds the intake air toward the front area and the rear area of the car chamber and aforementioned heater cores (13) and (23) were respectively arranged in aforementioned first ventilation trunk (11) and aforementioned second ventilation trunk (21).

3. The heating system for automobile described in Claim 1 or Claim 2 characterized by the fact that aforementioned first heat exchanger (3) and aforementioned second heat exchanger (6) were integrated together with liquid tank (4) and expansion valve (5).

Detailed explanation of the invention

[0001]

Industrial application field

The present invention relates to a heating system for automobile, which heats the hot water by utilizing the high-temperature high-pressure coolant in the refrigerating cycle.

[0002]

Prior art

For example, in some of the recent high class automobiles and one box cars with a relatively large space within the car chamber, an air conditioning system for automobile normally referred to as a dual air conditioner, which separately air conditions the front area (e.g., the front seat section) of the car chamber with a front unit and the rear area (e.g., the rear seat section such as the second seat, third seat, etc.) with a rear unit so that a comfortable air conditioned state can be achieved throughout the entire car chamber.

[0003]

This type of air conditioning system for automobile, utilizes a system wherein the front unit uses the engine cooling water as the heat source during a heating operation and the rear unit uses the high-temperature high-pressure coolant that was compressed according to a compressor as the heat source. Incidentally, this type of system is referred to as a heat pump type air conditioning system for automobile due to heating the car chamber by pumping up heat from the outside air of low temperature in the circulation process (refrigerating cycle) of the coolant.

[0004]

However, when carrying out a heating operation with this type of system, the temperature of the engine cooling water is low at the start when the outside temperature is low such as winter mornings and the increasing speed of the coolant temperature is also slow, hence it is difficult to achieve a state of warm air blowing out concurrent to start of driving, so-called immediate heating is insufficient, and there is a concern over even the heating performance becoming deficient. Also, in an one box car with a large space in the car chamber loaded with a diesel engine in particular, increase in the temperature of engine cooling water is slow in comparison with a car having a regular gasoline engine, and moreover, a wide space has to be heated hence there is a tendency for the immediate heating property and heating performance becoming deficient.

[0005]

Therefore, a heat pump type automobile air conditioning system has been developed, which was designed for high heating performance by heating the coolant utilizing the heat of the engine cooling water and using the coolant of high temperature obtained from increase in the enthalpy (e.g., refer to Patent Application Number Hei 7[1995]-271621).

[0006]

Problems to be solved by the invention

However, in the aforementioned heat pump type system, which heats the air by the front unit utilizing hot water and the rear unit utilizing the high-temperature high-pressure coolant, there are the following concerns.

[0007]

First, the rear unit generally decreases the discharge pressure by performing on/off control of the compressor when the compressor discharge pressure rises in order to protect the compressor during a refrigerating cycle, hence there is a fixed limit to the increase in the coolant temperature. Therefore, there is a fixed limit even in the temperature of the air heated according to heat exchange, namely, the temperature of the blowout air (e.g., when the reliability of the compressor is considered, only maximum air outlet temperature of about 60°C can be obtained).

[0008]

Second, the compressor is constantly operated not only during a heating (when the outside temperature is low) but even during a temperature control (when the outside temperature

is median) hence there is a concern over the fuel consumption dropping with respect to a regular heating system, which heats the air by exclusively utilizing hot water.

[0009]

Third, if by some chance the compressor, etc. fails, the rear unit will cease to function all together and the rear seat cannot be heated.

[0010]

Fourth, a separate heat source is used for the front unit and the rear unit (hot water for the former and high- temperature high-pressure coolant for the latter) hence it is not possible to improve the heating performance of the front unit and the rear unit concurrently with one means.

[0011]

Incidentally, in a cold region for example, there are many vehicles loaded only with a system that heats the car chamber (automobile heating system) as an air conditioning system and there was a great demand for such automobile heating system, hence development of an automobile heating system capable of solving the aforementioned concerns by focusing only on heating was necessary.

[0012]

The present invention was made by giving attention to the aforementioned problems in the heat pump type automobile air conditioning system currently being developed by the present applicant and the purpose thereof is to provide a totally new automobile heating system which solves all the aforementioned concerns while based on the existing hot water type heater.

[0013]

Means to solve the problems

In order to achieve the aforementioned objective, the invention according to Claim 1 is characterized by the fact that heater cores, which heat the intake air using the hot water for cooling the engine, were provided within the ventilation trunks, which feed the intake air toward the car chamber, a first heat exchanger, which heats the hot water flowing into the heater cores by utilizing the coolant discharged from the compressor composing a refrigerating cycle was provided between the hot water inlets of the heater cores and the hot water outlet of the aforementioned engine, and a second heat exchanger, which cools the hot water flowing out from the aforementioned heater cores by utilizing the coolant flowing out from the aforementioned

first heat exchanger was provided between the hot water outlets of the aforementioned heater cores and the hot water inlet of the aforementioned engine.

[0014]

In this invention, the hot water flowing into the heater cores from the engine is heated by utilizing the coolant discharged from the compressor, namely, according to heat exchange (radiation from the cycle) with the coolant of high-temperature high-pressure discharged from the compressor in the first heat exchanger and the temperature increases. As a consequence, hot water flows into the heater cores hence the air passing through the heater cores is heated to a higher temperature and the temperature of the blowout air increases. Also, the hot water returning to the engine from the heater cores is cooled in the second heat exchanger by utilizing the coolant flowing out from the first exchanger, namely, according to heat exchange (heat adsorption to the cycle) with the easily gasifying coolant of low-temperature low-pressure, which flowed out from the first heat exchanger, passed through a liquid tank, and was applied with adiabatic expansion in the expansion valve, and then returns to the engine. Namely, hot water is heated indirectly instead of air being heated directly by utilizing a high-temperature high-pressure coolant like in the so-called heat pump type heater. However, the temperature of the hot water flowing through the heater cores increases by heating the hot water hence the heating performance improves in comparison with the regular hot water type heater. Also, hot water that increases in temperature to almost 100°C is the heat source of the blowout air hence the heating performance improves in comparison with the heat pump type heater which has a fixed limit in the increase of the coolant temperature due to control of the compressor discharge pressure (Pd).

[0015]

Also, if the compressor is stopped, it operates as a regular hot water type heater, hence the compressor need not be operated constantly if composed to stop after the temperature of the hot water has increased sufficiently.

[0016]

Also, even if the compressor, etc. fails, it can operate as a regular hot water type heater even though heating of the hot water according to heat exchange with the coolant becomes impossible hence heating itself is still possible.

[0017]

The invention according to Claim 2 is characterized by the fact that in the aforementioned invention described in Claim 1, there is a first ventilation trunk and a second ventilation trunk, which respectively feeds the intake air toward the front area and the rear area of the car chamber and the aforementioned heater cores were respectively arranged in the aforementioned first ventilation trunk and the aforementioned second ventilation trunk.

[0018]

In this invention, both the front area (front side) and the rear area (rear side) of the car chamber are heated according to the hot water heated according to heat exchange with a coolant. As a consequence, the heating performance on the front side and the rear side can be improved concurrently with one means.

[0019]

The invention according to Claim 3 is characterized by the fact that in the invention described in Claim 1 or Claim 2, the aforementioned first heat exchanger and the aforementioned second heat exchanger, were integrated together with a liquid tank and an expansion valve.

[0020]

In this invention, the first heat exchanger, the second heat exchanger, the liquid tank, and the expansion valve are integrated and space efficiency is achieved. For example, mounting in the space under the floor of the vehicle as one unit is possible.

[0021]

Embodiments of the invention

Below, working examples of the present invention will be described using the drawings. Figure 1 is a schematic diagram showing a working example of a heating system for automobile related to the present invention. This heating system for automobile has front unit (10) and rear unit (20), which heat the inside and outside air taken in with an air blower and blows out towards the front seat and the rear seat in the car chamber.

[0022]

Both of these units (10) and (20) are respectively constituted by arranging air blowers (12), (22) and heater cores (13), (23) in order from the upstream side in the flow direction of air indicated with the stripped arrow within ventilation trunks (11) and (21), which are formed within a casing like in the conventional generic heating system for automobile. Hereinafter,

heater core (13) in front unit (10) will be referred to as front heater core and heater core (23) in rear unit (20) as rear heater core. Heater cores (13) and (23) are heat exchangers that heat the intake air by utilizing the cooling water (hot water) in engine (1). Though not shown in the figure, to be more specific, both units (10) and (20) are composed to have an intake unit and a heater unit in order from the upstream side and both are connected with a heater duct. An intake door and aforementioned air blowers (12) and (22) are arranged in the intake unit and an air mix door and aforementioned heater cores (13) and (23) are arranged in the heater unit. The air mix door is provided on the front faces of heater cores (13) and (23) and creates air of desired temperature at the downstream area of heater cores (13) and (23) by adjusting the ratio of the hot air that passed through heater cores (13) and (23) and the cold air (outside air) which bypassed heater cores (13) and (23) or prevents air from flowing through heater cores (13) and (23). Also, various air outlets to the front seat and the rear seat in the car chamber are respectively formed on the downstream side of heater cores (13) and (23) in the heater unit.

[0023]

Engine (1) and heater cores (13) and (23) in units (10) and (20) are connected respectively with a hot water piping so that the hot water flowing out from the engine is divided at midway and flows into heater cores (13) and (23) and the hot water flowing out from heater cores (13) and (23) is converged at midway and returns to engine (1). Also, though not shown in the figure, a valve is provided to the respective hot water inlet of heater cores (13) and (23) and hot water is introduced respectively to heater cores (13) and (23) from engine (1) by opening these valves.

[0024]

Also, compressor (2), which is rotated according to engine (1), condenser (3), which functions as the first heat exchanger and is capable of circulating hot water, liquid tank (4), expansion valve (5), and evaporator (6), which functions as the second heat exchanger and capable of circulating hot water are arranged on the outside section of both units (10) and (20). A refrigerating cycle is composed by connecting the compressor (2), condenser (3), liquid tank (4), expansion valve (5), and evaporator (6) with piping. For example, condenser (3) and evaporator (6) have the structure shown schematically in Figure 2 and a passage through which hot water flows and a passage through which a coolant flows are respectively formed on the respective inside part and composed for heat exchange to be carried out between the hot water and the coolant.

[0025]

Condenser (3) is provided between the (dividing point of) hot water inlets of heater cores (13) and (23) and the hot water outlet of engine (1) and has a function of heating the hot water flowing into heater cores (13) and (23) from engine (1) by utilizing the coolant discharged from compressor (2), namely, according to heat exchange (radiation from the cycle) with the high-temperature high-pressure coolant discharged from compressor (2)

[0026]

On the other hand, evaporator (6) is provided between the (converging point of) hot water outlets of heater cores (13) and (23) and the hot water inlet of engine (1) and has a function of cooling the hot water returning to engine (1) from heater cores (13) and (23) by utilizing the coolant flowing out from condenser (3), namely, according to heat exchange (heat absorption to the cycle) with the easily gasifying coolant of low-temperature and low-pressure, which flowed out from condenser (3), passed through liquid tank (4), and applied with adiabatic expansion in expansion valve (5).

[0027]

Next, the operation will be described. When both front unit (10) and rear unit (20) open the respective valve (not shown in the figure) during a heating, hot water flows respectively into front heater core (13) and rear heater core (23) from engine (1) via condenser (3). After flowing through the inside, the hot water flows out respectively from front heater core (13) and rear heater cores (23) and returns to engine (1) via evaporator (6). When compressor (2) is turned on at this time, a refrigerating cycle wherein the coolant circulates in the order of compressor (2) → condenser (3) → liquid tank (4) → expansion valve (5) → evaporator (6) → compressor (2) is established and condenser (3) and evaporator (6) function respectively. Namely, the coolant suctioned into compressor (2) in a superheated steam state is applied with adipabatic compression in compressor (2), becomes a high-temperature high-pressure gas coolant, and is discharged from compressor (2). This high-temperature high-pressure gas coolant is guided into condenser (3) and here radiates heat to the hot water that flowed in from engine (1), heats this hot water, and becomes a median-temperature high-pressure liquid coolant. Next, this liquid coolant is applied with adiabatic expansion by passing through expansion valve (5) via liquid tank (4), becomes an easily gasifying coolant of gas and liquid mixture, and then is guided to evaporator (6). Here, heat is absorbed from the hot water flowing out from heater cores (13) and (23), cools this hot water, isobaric expansion is continued while evaporating, and suctioned into compressor (2) again as superheated steam. Therefore, the hot water flowing into heater cores (13) and (23) from engine (1) increases in temperature by being heated according to heat

exchange with the high-temperature high-pressure gas coolant in condenser (3), hence hot water of high temperature flows into heater cores (13) and (23) and the heating performance of heater cores (13) and (23) improves. As a result, the air passing through heater cores (13) and (23) is heated to a higher temperature. Also, the hot water returning to engine (1) from heater cores (13) and (23) decreases in temperature by being cooled according to heat exchange with the easily gasifying coolant of low-temperature low-pressure in evaporator (6), hence the cooling effect of engine (1) is enhanced. Incidentally, if compressor (2) is turned off, the aforementioned refrigerating cycle is not established and functions as a regular hot water type heater that utilizes hot water.

[0028]

At this time, the air taken into front unit (10) and rear unit (20) according to air blowers (12) and (22) is heated in respective heater cores (13) and (23) and then flows downward and blows out to the front seat and the rear seat in the car chamber from the predetermined air outlets. At this time, the front seat and the rear seat are heated or temperature controlled independently by adjusting the opening of each air mix door (not shown in the figure).

[0029]

Also, after the temperature of the hot water increases sufficiently, the heat pump system is stopped by turning off compressor (2) and is operated as a regular hot water type heater so that the hot water is not heated unnecessarily.

[0030]

Incidentally, it is possible to add a control at the initial stage of the heating so that air from the air mix doors (not shown in the figure) does not pass through heater cores (13) and (23) until the temperature of the blowout air reaches a certain temperature.

[0031]

Also, when controlling the temperature, the temperature of the air outlets is regulated by adjusting the opening of the air mix doors (not shown in the figure) and adjusting the mix ratio of the hot air heated in heater cores (13) and (23) and the cold air (outside air) that bypassed heater cores (13) and (23).

[0032]

Also, when it is not necessary to carry out a heating, valves (not shown in the figure) are closed to prevent hot water from being introduced into heater cores (13) and (23).

[0033]

Therefore, according to the present working example, hot water is heated indirectly instead of heating the air directly by utilizing a high-temperature high-pressure coolant like in the so-called heat pump type heater. However, the temperature of the hot water flowing through heater cores (13) and (23) increases by heating the hot water with the coolant, hence the heating performance improves in comparison with the regular hot water type heater. Also, hot water capable of increasing to a temperature of almost 100°C is used as the heat source for the blowout air hence the heating performance improves even when compared with the heat pump type heater, which has a fixed limit in increase of the coolant temperature due to control of the compressor discharge pressure (Pd) as was described above. For example, maximum air outlet temperature of over 70°C was obtained according to testing.

[0034]

Also, if compressor (2) is turned off, it operates as a regular hot water type heater, hence if it is composed to stop compressor (2) after the hot water temperature had increased sufficiently, it is not necessary to constantly operate compressor (2) and can achieve improvement in the fuel consumption.

[0035]

Furthermore, even if compressor (2), etc. fails, it still operates as a regular hot water type heater even if heating of the hot water according to heat exchange with the coolant becomes impossible, hence heating itself is still possible. Namely, recovery is possible when compressor (2) and the system fail.

[0036]

Also, both front unit (10) and rear unit (20) can be heated with the hot water, which was heated according to heat exchange with a coolant hence improvement in the heating performance can be achieved concurrently on both the front side and the rear side.

[0037]

Incidentally, it is preferable to integrate condenser (3), evaporator (6), liquid tank (4), and expansion valve (5) in order to achieve enhancement in the space efficiency and reduction in cost. Figure 3 is a plan view showing an example of such integrated unit and Figure 4 is a typical diagram showing the structure of this unit. Here, Figure 4(A) shows the structure for flow of the hot water and Figure 4(B) shows the structure for flow of the coolant.

[0038]

In the figures, (30) is the integrated unit, (31) the condenser part corresponding to condenser (3), (32) the liquid tank part corresponding to liquid tank (4), and (33) the evaporator part corresponding to evaporator (6). Expansion valve (5) is attached to evaporator part (33) of unit (30).

[0039]

When condenser (3), evaporator (6), liquid tank (4), and expansion valve (5) are integrated as described above, it can be mounted as one unit in the space under the floor of the vehicle, space saving is possible, and cost reduction can also be achieved. Also, liquid tank (4) and evaporator (6) were integrated and liquid tank part (32) and evaporator part (33) were arranged to make contact hence heat exchange is carried out between the two and the performance is improved.

[0040]

Effects of the invention

As described above, according to the invention described in Claim 1, hot water is heated utilizing a high-temperature high-pressure coolant hence the temperature of the hot water flowing through the heater cores increases and the heating performance improves. Also, even if the compressor is stopped, it functions as a regular hot water type heater hence it is not necessary to constantly operate the compressor and improvement in fuel consumption can be achieved. Furthermore, even if the compressor, etc. fails, it can continue to operate as a regular hot water type heater hence the heating itself is still possible.

[0041]

According to the invention described in Claim 2, in addition to the aforementioned effects of the invention described in Claim 1, heating is possible on both the front side and the rear side of the car chamber with the hot water heated according to heat exchange with the coolant hence improvement in the heating performance can be achieved concurrently on the front side and the rear side with one means.

[0042]

According to the invention described in Claim 3, in addition to the aforementioned effects of the invention described in Claim 1 and Claim 2, enhancement in the space efficiency

and reduction in the cost can be achieved since the first heat exchanger, the second heat exchanger, the liquid tank, and the expansion valve were integrated.

Brief description of the drawings

Figure 1 is a schematic diagram showing a working example of the heating system for automobile related to the present invention.

Figure 2 is a plan view showing the structure of the condenser and the evaporator shown in Figure 1.

Figure 3 is a plan view showing an example of an integrated unit

Figure 4 is a typical diagram showing the structure of said unit.

Explanation of reference numbers in the figures

1	Engine
2	Compressor
3	Condenser (first heater exchanger)
4	Liquid tank
5	Expansion valve
6	Evaporator (second heater exchanger)
11, 21	Ventilation trunks
13, 23	Heater cores
30	Integrated unit

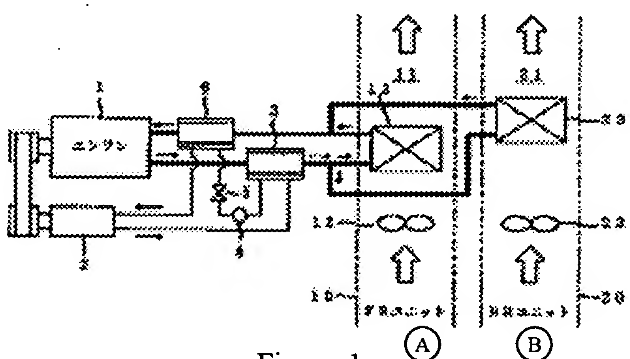

Figure 1

Key: A FR unit
B RR unit

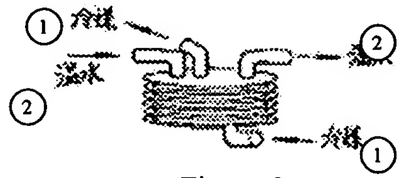

Figure 2

Key 1 Coolant
2 Hot water

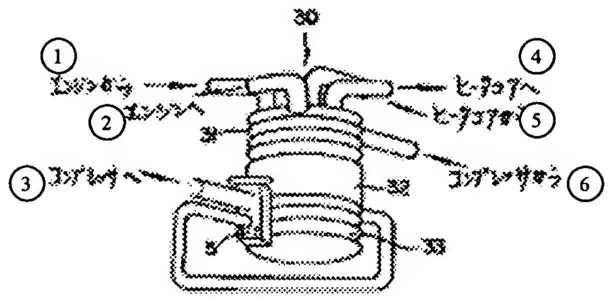

Figure 3

Key 1 From the engine
2 To the engine
3 To the compressor
4 To the heater core
5 From the heater core
6 From the compressor

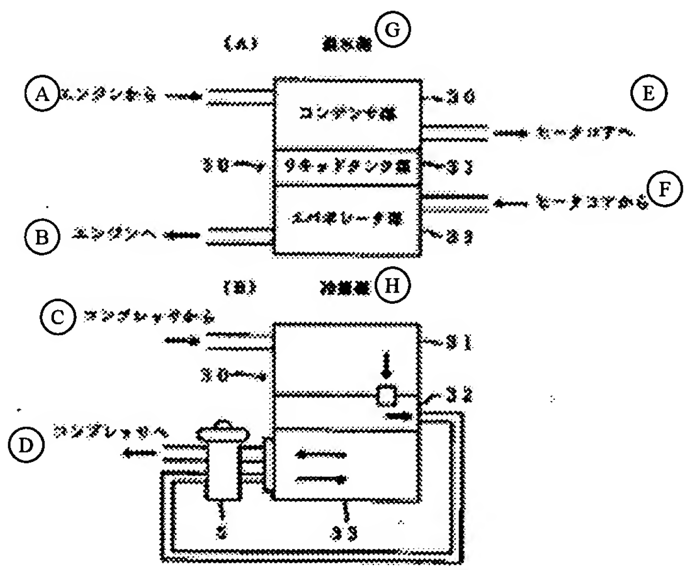

Figure 4

Key:	A	From the engine
	B	To the engine
	C	From the compressor
	D	To the compressor
	E	To the heater core
	F	From the heater core
	G	Hot water side
	H	Coolant side
	30	Condenser part
	31	Liquid tank part
	35	Evaporator part

copy of FR 2288278
with translation
for Appellant

③

RÉPUBLIQUE FRANÇAISE

INSTITUT NATIONAL
DE LA PROPRIÉTÉ INDUSTRIELLE

PARIS

(11) N° de publication : **2 288 278**
(A n'utiliser que pour les commandes de reproduction).

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DEMANDE
DE BREVET D'INVENTION

(21) **N° 74 35075**

(54) **Système à pompe de chaleur pour conditionnement de l'intérieur de bâtiments.**

(51) Classification internationale (Int. Cl.²). F 25 B 29/00; F 24 F 3/08.

(22) Date de dépôt .... 18 octobre 1974, à 14 h 15 mn.

(33) (32) (31) Priorité revendiquée :

(41) Date de la mise à la disposition du public de la demande .... B.O.P.I. — «Listes» n. 20 du 14-5-1976.

(71) Déposant : Société dite : BERGEON ET CIE, résidant en France.

(72) Invention de : Jean Patry.

(73) Titulaire : *Idem* (71)

(74) Mandataire : Massalski, Barnay et Grucy, Conseils en brevets d'invention.

Le se teur technique d l'inventi n est c lui du conditionn ment de l'intéri ur des bâtiments.

Il st déjà connu d'utilis r, pour l trait ment de l'atmosphère intérieure des bâtiments les plus divers, des pompes de chaleur. Généralement les systèmes proposés ne sont valables que pour le chauffage ou que pour la réfrigération. Les systèmes mixtes sont généralement complexes et nécessitent de nombreuses manoeuvres de vannes en cas d'inversion , c'est-à-dire le passage du cycle de chauffage au cycle de réfrigération et réciproquement.

Le but de la présente invention est principalement de simplifier de telles installations mixtes et leur manoeuvre.

heat pump

L'invention comprend à cet effet, un système à pompe de chaleur, comprenant un compresseur de fluide frigorigène, du genre de l'un de ceux qui sont connus sous le nom de "Fréon", associé à un évaporateur et un condenseur, respectivement source froide et source chaude et un réseau de fluide caloporteur, en général de l'eau, caractérisé par le fait que ledit réseau comport une pluralité de mailles, l'une, boucle courte chaude traversant ledit condenseur et un échangeur de chauffage, l'autre, boucle courte froide, traversant ledit évaporateur et un échangeur de refrigération, ces deux boucles courtes étant réunies en aval de leurs échangeurs et en amont par des liaisons, entre lesquelles s'étend une ligne sur laquelle est interposé un échangeur de récupération, des tiroirs à trois voies étant disposés aux jonctions entre liaison amont et voies d'arrivée et de départ des boucles courtes respectivement avant et après ces tiroirs.

Dans une forme d'exécution avantageuse, la commande de ces tiroirs est assurée, en cascade, à partir d'un régulateur thermométrique principal ayant son détecteur placé au contact de la boucle courte chaude, en amont de l'éehangeur chauffant correspondant. Il est également possible d'associer un régulateur d'énergie d'alimentation du compresseur, avec un détecteur au contact de la boucle courte de réfrigération, en amont de l'échangeur réfrigérant.

De même, les échangeurs de boucles courtes peuvent être pourvus de by-pass commandés par vannes à deux voies et régulateurs thermostatiques.

L régulat ur principal peut être à d ux entré s, dont l'autre st s nsible à la température extérieur .

Il y a avantage à prévoir sur l s boucles courtes d s pompes séparé s d'impulsion pour le fluid ‾caloporteur, en amont des condenseur t évaporat ur r spectivement.

Il y a également avantage à brancher en parallèle, dans la boucle courte de chauffage, en amont de l'échangeur correspondant, un générateur, accumulateur ou autre réserve d'eau, susceptible de recevoir en cas de besoin une énergie thermique d'appoint. De même, sur la ligne de récupération peut être branchée une source frigorifique d'appoint.

De même, l'échangeur de récupération peut être parcouru par l'air extrait de l'immeuble en cause.

Enfin, cet échangeur de récupération peut être disposé, au moins pour partie, dans un fluide à haute capacité thermique et à température aussi constante que possible, comme l'eau d'une nappe phréatique.

La description qui va suivre, en regard du dessin annexé à titre d'exemple non limitatif, permettra de bien comprendre comment l'invention peut être mise en pratique.

La figure unique représente un schéma d'une telle installation.

Dans l'ensemble représenté, une installation de conditionnement de locaux d'habitation comprend tout d'abord un circuit à pompe de chaleur comportant un compresseur Co entraîné par un moteur convenable, débitant par une tubulure 1 un fluide frigorigène et calorigène, du genre connu sous le nom de "Fréon" plus particulièrement dans un condenseur C, avec retour par une conduite 2 sur laquelle est interposé un détendeur D à un évaporateur E, duquel part une conduite 3 faisant retour à l'admission du compresseur Co.

Un réseau de fluide caloporteur, le plus généralement de l'eau, en raison de son prix et de sa capacité calorifique, comprend une pompe à eau chaude Pc dont l'aspiration est reliée par une tubulure à un tiroir ou vanne 4 à trois voies 4a, 4b et 4c, plus spécialement à la voie 4a. Le refoulement de la pompe à eau chaude Pc est relié à une tubulure 5 qui alimente une série d'échangeurs, ou au moins un échangeur général ou batterie Bc chaude, au travers d'un autre échangeur contenu dans le condenseur C. A l'entrée de la batterie Bc est disposé un tiroir 6 à d ux voies 6a et 6b dont la voie 6a commande l'entrée dans la batterie Bc et la voie 6b command l'entrée d'une tubulure 7 de

by-pass contournant ladit batteri et se raccordant à une tubulure 8 de sortie de batt ri .

Cette tubulure 8 se branche sur une tubulure 9 de retour à la voie 4b du tiroir 4 pour retour à l'admission de pompe Pc.

De même, du côté de l'évaporateur E, le réseau à fluide caloporteur comporte devant cet évaporateur E une pompe à eau glacée Pg qui est reliée par son refoulement à une tubulure 10 traversant l'échangeur intérieur à cet évaporateur puis aboutissant à un tiroir 11 à deux voies 11a et 11b; par la voie 11a est alimentée une batterie BF froide. La voie 11b est à l'entrée d'un by-pass 12 rejoignant la tubulure 13 de sortie de batterie BF. Cette tubulure 13 est branchée sur une tubulure 14 de retour à l'aspiration de la pompe Pg au travers d'un second tiroir à trois voies 15, par une voie 15b, puis une voie 15a. La voie 15c est directement réunie par une tubulure 16 à la voie 4c du tiroir 4. Une tubulure 17 réunit directement les tubulures 8 et 13. Une tubulure 18 réunit les tubulures 16 et 17 et sur cette tubulure 18 est disposée une batterie de réchauffage ou de refroidissement BR. Cette batterie peut être disposée dans le courant d'air sortant, après utilisation à l'intérieur de l'immeuble en cause.

La régulation de cette ensemble est confiée à un régulateur thermique RT convenablement alimenté par raccordement sur un source de courant alternatif, offrant un détecteur d_1 au contact avec la tubulure 5 d'entrée à la batterie BC et dont la commande agit en cascade sur les tiroirs 4 et 15, c'est-à-dire provoque la manoeuvre d'un tiroir lorsque l'autre a achevé son mouvement.

Un autre régulateur énergétique RE alimenté de même, possède un détecteur thermique d_2 au contact de la tubulure 10 de sortie de la pompe à eau glacée Pg et agit sur l'alimentation du compresseur Co.

Il est possible de brancher en parallèle sur la tubulur 5 un dispositif auxiliaire qui peut être un échangeur A thermique alimenté par une énergie extérieure - combustible par exemple - un accumulateur thermique ou les deux à la fois. De même sur la tubulure 18 peut être branché en dérivation un échangeur F à groupe frigorigène. Ces dérivations comportent des vannes d'isolement et une vanne d'interruption de la liaison directe.

Le fonctionnement d cet ns mbl est le suivant :

Le régulateur RT, réglé par xempl sur 50°C est d stiné à agir sur les tiroirs 4 et 15 en vue d maintenir une température

constante à la batterie BC. Ce régulateur peut être avantag usement à deux entrées, afin d définir, par préaffichag , une températur de sortie d' au d condenseur C qui soit fonction de la température extérieure à l'immeuble en cause.

Le régulateur RE est dans la majorité des cas, formé par un thermostat à plusieurs étages dont le bulbe de mesure sur l'eau glacée de la tubulure 10 agit sur la puissance développée sur le compresseur Co.

Des régulateurs thermostatiques sont associés aux deux tiroirs à deux entrées 6 et 11, régulateurs pouvant être intégrés auxdits tiroirs, pour maintenir constantes les températures régnant dans les batteries BC et BF.

Dans une phase correspondant aux besoins calorifiques maximals du système, la ou les batteries BC consomment des calories et la température de l'eau dans le circuit de condenseur C a tendance à s'abaisser. Le régulateur RT met en positions les tiroirs 4 et 15 de telle sorte que :

- dans le tiroir 4, l'ouverture est réalisée entre les voies 4<u>b</u> et 4<u>a</u>, la voie 4<u>c</u> étant fermée
- dans le tiroir 15, l'ouverture est réalisée entre les voies 15<u>c</u> et 15<u>a</u>, la voie 15<u>b</u> étant fermée.

L'eau chaude produite au condenseur C sous l'action du compresseur Co parcourt le circuit :

PC → C → BC → voies 4<u>b</u>, 4<u>a</u> dans le tiroir 4 (la voie 4<u>c</u> étant obturée) → retour à PC.

L'eau glacée produite à l'évaporateur E parcourt le circuit :

Pg → E → BF → BR → voies 15<u>c</u>, 15<u>a</u> dans le tiroir 15 (la voie 15<u>b</u> étant obturée) → retour à Pg.

Les frigories consommées par la batteries BF et la batterie BR sont transférées par l'action du compresseur Co au condenseur C, augmentées du travail nécessaire à ce transfert.

Remarque doit être faite qu'aux très basses températures extérieures, si le bilan énergétique est tel que les calories engendrées dans ce système sont insuffisantes, on peut mettre en action le dispositif A qui peut être un échangeur recevant de l'énergie calorifique extérieure, chauffe-eau à combustible ou accumulateur d'eau à énergie électrique de nuit, ou autre. On obtient ainsi le complément calorifique voulu dans de telles circonstances.

Il est à remarquer que la batterie BR, parcourue par l'air expulsé du bâtiment en cause, récupère ainsi par

consommation des frigories dans l circuit d'eau glacé ainsi établi , un grande parti d chaleur, l système en caus n'ayant alors à fournir que la compensation d s pert s, aux rendements près, inhérentes à l'édifice lui-même et à d'autres facteurs tels que diminution de fréquentation ou autres.

Dans une autre phase correspondant à des besoins calorifiques réduits du système et ce, jusqu'à l'approche de l'équilibre thermique moyen naturel de celui-ci, la consommation de frigories du circuit d'eau glacée est strictement adaptée aux besoins. En effet, la vanne thermostatique 11 module le taux de passage de cette eau dans la ou les batteries BF et le régulateur RE énergétique module l'action du compresseur Co.

Dans une autre phase correspondant à des besoins calorifiques nuls du système et donc pour son point d'équilibre thermique moyen naturel, le régulateur thermique RT fait basculer le tiroir 15 pour fermer la voie 15c et établir la communication directe 15b, 15a isolant ainsi la batterie BR, puisque le tiroir 4 n'a pas changé de position.

L'eau chaude produite au condenseur C parcourt le circuit : PC ⟶ C ⟶ BC ⟶ voies 4b, 4a (la voie 4c demeurant obturée) ⟶ retour à PC.

L'eau glacée produite à l'évaporateur E parcourt le circuit :

Pg ⟶ E ⟶ BF ⟶ voies 15b (ouverte), voie 15a, retour à Pg.

Le compresseur, tout en assurant les besoins éventuels de consommation de frigories des batteries froides BF - réglés par la vanne 11, est modulé à une puissance mécanique minimale par le régulateur énergétique RE. Dès que cet équilibre thermique moyen naturel du système est atteint, ceci correspond au fait qu'il est en léger excès de calories et il est alors possible d'absorber ces calories dans un chauffe-eau tel que A, qui au lieu de consommateur d'énergie externe devient producteur d'eau chaude, utilisable sur place pour les besoins sanitaires et autres.

La dernière phase à envisager ici est celle des besoins frigorifiques du système. Dans cette phase, le régulateur thermique RT agit sur le tiroir 4 pour mettre en communication les voies 4a et 4c en fermant la voie 4b. Le tiroir 15 maintient établi le circuit court par la tubulure 14 (voi 15c obturée, voi s 15b et 15a en communication) alors que l tiroir 4 établit l circuit long

pour l'eau chaude, la faisant ainsi passer par la batteri BR qui joue alors le rôle d'un aéro condenseur, évacuant l'excès d calori s à l'extéri ur, n rechauffant ainsi dès sa sortie l'air expulsé de l'édifice où le système en cause est installé.

Il est à souligner que la batterie BR pourrait être agencée autrement que sur le parcours de l'air sortant de l'édifice et qu'elle pourrait au moins partiellement être plongée dans un autre milieu, tel qu'une nappe phréatique à haute possibilité d'échange et à température quasiment constante, pour des raisons d'économie d'investissement notamment, mais alors à moindre récupération des calories ou frigories développées à l'intérieur du bâtiment.

En conclusion, l'installation décrite comprend une boucle hydraulique unique, à configuration modulable, faisant naître les avantages ci-après :

- transfert permanent d'un lieu à un autre d'un édific pourvu d'une telle installation des frigories ou calories consommées : des frigories consommées sur BF ou BR, par exemple, entraînent l'apparition d'un nombre sensiblement égal de calories à disposition sur BC, avec augmentation due à l'énergie mécanique nécessaire au transfert, d'où consommation énergétique globale réduite;

- mise à disposition, été comme hiver (sans aucune discontinuité de demi-saison, due à une inversion de cycle, comme dans les installations traditionnelles) de toute la puissance potentielle calorifique ou frigorifique;

- récupération totale jusqu'à une température extérieure de +6°C de toute l'énergie nécessaire au chauffage de l'air neuf alimentant l'édifice en cause;

- récupération des calories de l'air extérieur introduit à titre d'air neuf nécessaire à combattre les déperditions pour toutes les températures extérieures supérieures à +6°C.

Cette limite de +6°C est fixée par la nature du fluide caloporteur choisi dans le cas de cette valeur numérique comme étant de l'eau, pour éviter tout givrage et tout gel.

- régulation simple, à l'aide du régulateur principal RT qui agit sur la circulation du fluide caloporteur en répartition dans les diverses maill s du réseau; avec adjonction supplémentaire éventuelle des régulateurs thermostatiques de batteries et régulateur d'én rgi RE donnée au compr sseur Co;

- possibilité d'isoler complèt ment d s échangeurs de condens ur t d'évaporation la batteri d récupération BR surtout dans le cas d'utilisation d'une nappe phréatique, ce qui élimine des deux premiers échangeurs les risques d'encrassement ;

- réduction de la température de bulbe humide d'entrée d'air en tour de refroidissement. Cas dans lequel l'échangeur BR est ainsi utilisé - par envoi de l'air extrait du bâtiment sur cet échangeur de récupération en vue du refroidissement de l'eau du circuit traversant le condenseur C, en régime de fonctionnement "d'été";

- possibilité de fourniture d'eau chaude par extraction des calories de l'ambiance intérieure au bâtiment, à usage sanitaire notamment, en régime d'été.

Il va de soi que, sans sortir du cadre de l'invention, il est possible d'apporter des modifications aux formes d'exécution qui viennent d'être décrites.

-REVENDICATIONS-

1.- Système à pompe de chaleur compr nant un compresseur à fluide frigorigène associé à un évaporat ur t un cond nseur, respectivement source froide et source chaude dans un réseau à fluide caloporteur, caractérisé par le fait que ledit réseau comporte une pluralité de mailles, l'une, boucle courte chaude traversant ledit condenseur et un échangeur au moins de chauffage, l'autre ,boucle courte froide, traversant ledit évaporateur et au moins un échangeur de réfrigération, ces deux boucles étant retirées, en aval de leurs échangeurs et en amont des condenseurs d'évaporateur par des liaisons, entre lesquelles s'étend une ligne de récupération sur laquelle est interposé un échangeur correspondant au moins, des tiroirs à trois voies étant placés entre liaison en amont du condenseur et de l'évaporateur et boucles courtes correspondantes.

2.- Système selon la revendication 1, caractérisé par l fait que la commande de ces tiroirs à trois voies est assurée, en cascade, par un régulateur principal dont l'élément détecteur thermométrique est placé sur la boucle courte chaude, en amont d l'échangeur chauffant correspondant.

3.- Système selon la revendication 1 ou la revendication 2, caractérisé par le fait qu'un régulateur d'énergie secondaire est pourvu d'un élément détecteur thermométrique placé sur la boucle courte froide, en amont de l'échangeur correspondant et d nt l'élément de commande dose l'énergie fournie audit compresseur.

4.- Système selon l'une quelconque des revendications 1 à 3, caractérisé par le fait que les échangeurs des boucles chaude et froide sont pourvus de by-pass à ouverture réglée par vannes à deux voies actionnées par régulateurs thermostatiques, sensibles aux températures d'entrée de fluide caloporteur dans lesdits échangeurs.

5.- Système selon l'une quelconque des revendications 1 à 4, caractérisé par le fait que ledit régulateur principal est à deux entrées dont l'une est sur ladite boucle chaude et l'autre sensible à la température extérieure.

6.- Système selon l'une quelconque des revendications 1 à 5, caractérisé par le fait que les boucles courtes comportent ~~chacune~~ une pomp cimulatoire du fluid caloporteur corr spondant, pompes situées en amont des échangeurs respectifs.

7.- Système s lon l'une quelc nqu des r v ndications 1 à 6, caractérisé par le fait que la boucl courte chaude comporte, en amont de l'échangeur correspondant, un accumulat ur, un générateur, ou les deux à la fois, susceptible de recevoir un appoint d'énergie thermique extérieure.

8.- Système selon l'une quelconque des revendications 1 à 7, caractérisé par le fait que la ligne de récupération traverse en outre un élément formant source frigorigène d'appoint.

9.- Système selon l'une quelconque des revendications 1 à 8, caractérisé par le fait que l'échangeur de récupération est exposé à l'air extrait de l'immeuble.

10.- Système selon l'une quelconque des revendications 1 à 9, caractérisé par le fait que ledit échangeur de récupération est exposé, au moins pour partie, à un fluide à haute capacité thermique de température annuelle sensiblement constante.

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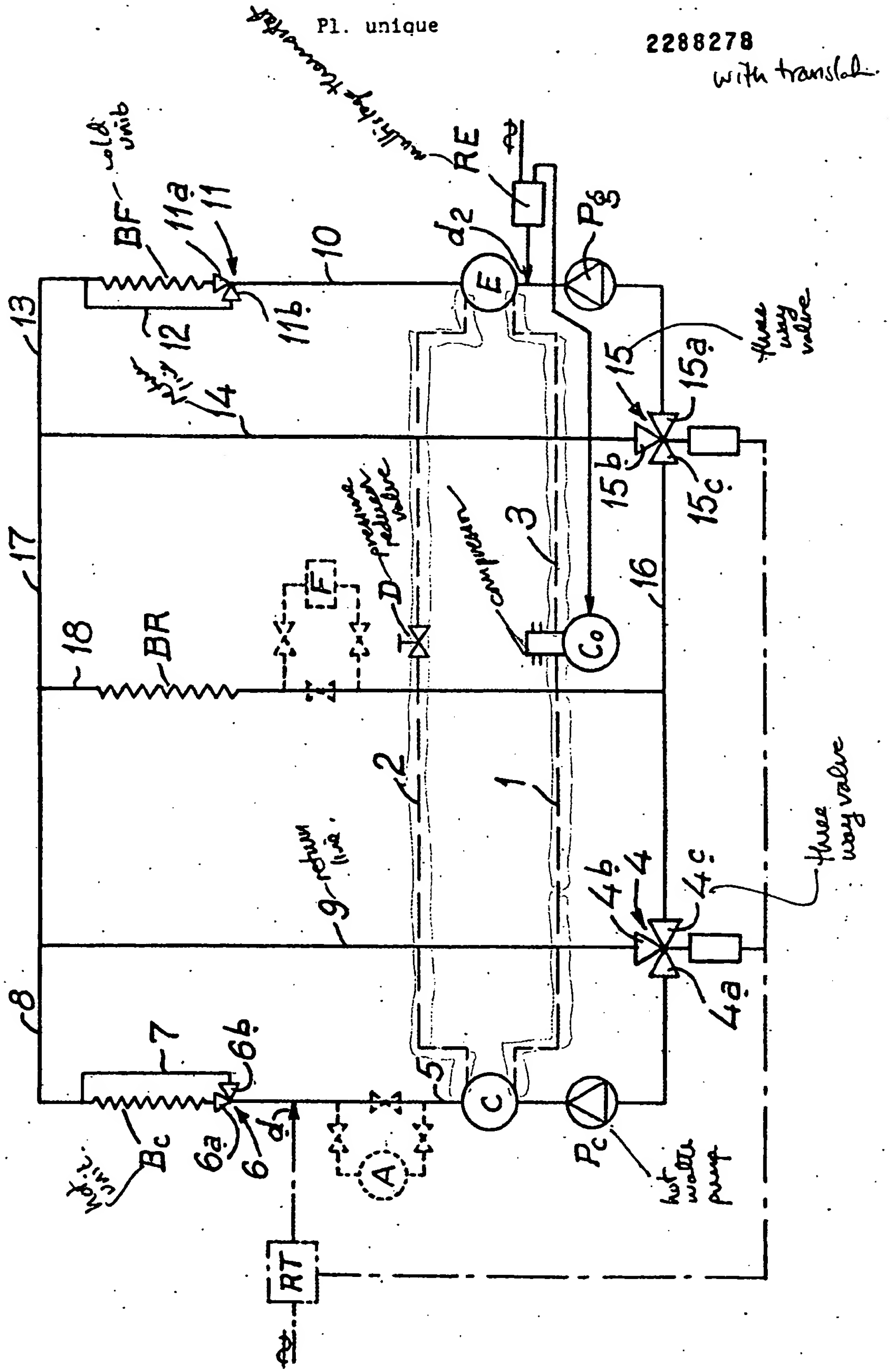


DERWENT-ACC-NO: 1976-H2863X

DERWENT-WEEK: 197633

TITLE: Heat pump for heating and cooling - has hot and cold water branches and three-way valves operated in cascade

PATENT-ASSIGNEE: BERGEON & CIE[BERGN]

PRIORITY-DATA: 1974FR-0035075 (October 18, 1974)

PATENT-FAMILY:

PUB-NO	PUB-DATE	LANGUAGE	PAGES	MAIN-IPC
FR 2288278 A	June 18, 1976	N/A	000	N/A

INT-CL (IPC): F24F003/08, F25B029/00

ABSTRACTED-PUB-NO: FR 2288278A

BASIC-ABSTRACT:

The heat pump for heating and cooling purposes has a compressor (Co), a condenser (c), an evaporator (L) and a pressure reduction valve (D), around which the refrigerant is circulated, and a hot water circuit with a pump (Pc) which passes the water through a three-way valve (4) and heat exchangers (Bc), with a by-pass line (7) across them and a return line (9), and also through a cold circuit with another three-way valve (15) in series with the evaporator (E) and a cold unit (BF). It has also a by-pass line (12) and a return line (14). The whole is controlled by a heat regulator connected to the hot section and a multi-stage thermostat (RE) at the cold section, operated by the compressor. The two three-way valves are operated in cascade.

TITLE-TERMS: HEAT PUMP HEAT COOLING HOT COLD WATER BRANCH THREE WAY VALVE OPERATE CASCADE

DERWENT-CLASS: Q74 Q75

PTO 05-4658 French Patent Application No.: 2 288 278

HEAT PUMP SYSTEM FOR [AIR] CONDITIONING OF THE INTERIOR OF BUILDINGS

Jean Patry

UNITED STATES PATENT AND TRADEMARK OFFICE
WASHINGTON, D.C. JULY 2005
TRANSLATED BY THE MCELROY TRANSLATION COMPANY

FRENCH REPUBLIC
NATIONAL INSTITUTE OF INDUSTRIAL PROPERTY
PATENT NO. 2 288 278

International classification[2]:	F 25 B 29/00 F 24 F 3/08
Filing number:	74 35075
Filing date:	October 18, 1974, 2:15 p.m.
Date of public access to the application:	B.O.P.I. – "Listes" No. 20, May 14, 1976

HEAT PUMP SYSTEM FOR [AIR] CONDITIONING OF THE INTERIOR OF BUILDINGS

[Système à pompe de chaleur pour conditionnement de l'intérieur de bâtiments.]

Inventor:	Jean Patry
Applicant:	Company known as: Bergeon et Cie, residing in France

The technical sector of the invention is that of [air] conditioning of the interior of buildings.

Already known for treatment of the interior of a wide variety of buildings is the use of heat pumps. Generally, the proposed systems are good only for heating or only for cooling. The mixed systems are generally complex and require numerous valve maneuvers in case of reversal, that is to say switching from the heating cycle to the cooling cycle and vice versa.

The aim of the present invention is mainly to simplify such mixed installations and their maneuvering.

To this effect, the invention entails a heat pump system which has a compressor with liquid refrigerant of the type known as "Freon," associated with an evaporator and a condenser, which are respectively the source of cold and the source of heat, and a network of liquid coolant, generally water, characterized by the fact that said network has a number of loops, one being a short loop for heat passing through said condenser and a

heating exchanger, the other being a short loop for cold passing through said evaporator and a cooling exchanger, these two short loops being joined downstream from their exchangers and upstream by connections, between which a line extends on which a recovery exchanger is inserted, three-way slide valves being arranged at the junctions between upstream connection and intake and exit ways of the short loops respectively before and after these slide valves.

In an advantageous form of execution, the control of these slide valves is ensured, in a series, by means of a main thermometric regulator whose detector is put in contact with the short loop for heat, upstream from the corresponding heating exchanger. It is also possible to associate a regulator of energy for supply of the compressor, with a detector in contact with the short loop for cooling, upstream from the cooling exchanger.

Likewise, the exchangers of short loops can be provided with bypasses controlled by two-way valves and thermostatic regulators.

The main regulator can have two intakes, the other of which is sensitive to the exterior temperature.

It is advantageous to provide separate impeller pumps on the short loops for the liquid coolant, upstream from the condenser and the evaporator respectively.

It is also advantageous to connect in parallel, in the short loop for heating, upstream from the corresponding exchanger, a generator, accumulator or other reserve of water, capable of receiving if necessary an additional thermal energy. Likewise, on the recovery line, an additional refrigerating source can be connected.

Likewise, the recovery exchanger can be run through by the air extracted from the building in consideration.

Finally, this recovery exchanger can be arranged, at least in part, in a fluid with a high thermal capacity and having a temperature which is as constant as possible, such as the water of a water table.

The following description, with regard to the appended drawing as a non-limiting example, will enable one to understand how the invention can be put in practice.

The single figure represents a diagram of such an installation.

In the assembly represented, an installation for [air] conditioning of residential dwellings first of all entails a heat pump circuit which has compressor Co driven by a suitable motor, which, through hose 1, delivers a refrigerating and heat-producing liquid of the type known by the name of "Freon" more particularly to condenser C, with return, through pipe 2 on which pressure release valve 2 is inserted, to evaporator E, from which pipe 3 leaves returning to the intake of compressor Co.

A network of liquid coolant, most generally water because of its price and its thermal capacity, includes hot water pump Pc whose suction is connected through a hose to slide valve or valve 4 with three ways 4a, 4b and 4c, and more particularly to way 4a. The delivery of hot water pump Pc is connected to hose 5 which supplies a series of exchangers, or at least one general exchanger or battery Bc for heat, through another exchanger contained in condenser C. Arranged at the intake of battery Bc is slide valve 6 with two ways 6a and 6b of which way 6a controls the intake into battery Bc and way 6b controls the intake of bypass hose 7 which circumvents said battery and is connected to battery outlet hose 8.

This hose 8 is connected to hose 9 for return to way 4b of slide valve 4 for return to the intake of pump Pc.

Likewise, on the evaporator E side, in front of this evaporator E, the liquid coolant network has chilled water pump Pg which is connected by its delivery to hose 10 which passes through the exchanger inside this evaporator and then ends at slide valve 11 with two ways 11a and 11b; cold battery BF is supplied through way 11a. Way 11b is at the entrance of bypass 12 which rejoins battery BF outlet hose 13. This hose 13 is connected to hose 14 for return to the suction of pump Pg through second three-way slide valve 15, through way 15b, then way 15a. Way 15c is directly connected by hose 16 to way 4c of slide valve 4. Hose 17 directly connects hoses 8 and 13. Hose 18 connects hoses 16 and 17, and on this hose 18, heating or cooling battery BR is arranged. This battery can be arranged in the stream of air exiting after use inside the building in consideration.

The regulation of this assembly is in the hands of thermal regulator RT suitably supplied by connection to a source of alternating current, offering detector d_1 in contact with hose 5 going into battery BC and the control of which acts in a series on slide valves 4 and 15, that is to say brings about the maneuvering of one slide valve when the other has completed its movement.

Another energy regulator RE supplied in a like manner has thermal detector d_2 in contact with hose 10 coming out of chilled water pump Pg and acts on the supply of compressor Co.

It is possible to connect in parallel on hose 5 an auxiliary device which can be thermal exchanger A supplied by exterior energy – fuel, for example – a thermal accumulator or both at the same time. Likewise, on hose 18, exchanger F with a cold-producing unit can be connected in the form of a bypass. These bypasses have isolation valves and a valve for interruption of the direct connection.

The functioning of this assembly is the following:

Regulator RF, set, for example, on 50°C, is intended for acting on slide valves 4 and 15 for the purpose of maintaining a constant temperature in battery BC. This regulator can advantageously have two ways, in order to define, by pre-displaying [misprint ? of préchauffage = preheating], a temperature of water coming out of condenser C which is a function of the temperature outside the building in consideration.

Regulator RE is in the majority of cases formed by a thermostat with several stages whose measuring bulb on the chilled water of hose 10 acts on the power developed in compressor Co.

Thermostatic regulators are associated with the two slide vales 6 and 11 with two intakes, regulators which can be integrated in said slide valves, in order to maintain the temperatures predominating in batteries BC and BF constant.

In a phase corresponding to the maximum heat needs of the system, battery or batteries BC consume calories, and the temperature of the water in the circuit of condenser C has a tendency to go down. Regulator RT puts slide valves 4 and 15 in position so that:

- in slide valve 4, the opening is brought about between ways 4b and 4a, way 4c being closed;

- in slide valve 15, the opening is brought about between ways 15c and 15a, way 15b being closed.

The hot water produced in condenser C under the action of compressor Co travels the circuit:

PC → C → BC → ways 4b, 4a in slide valve 4 (way 4c being closed) → return to PC.

The chilled water produced in evaporator E travels the circuit:

Pg → E → BF → BR → ways 15c, 15a in slide valve 15 (way 15b being closed) – return to Pg.

The negative kilogram calories consumed by batteries BF and battery BR are transferred by the action of compressor Co to condenser C, increased by the work necessary for this transfer.

It should be noted that at very low exterior temperatures, if the energy balance is such that the calories generated in this system are insufficient, it is possible to activate device A, which can be an exchanger which receives the exterior heat energy, a water heater using fuel or a water accumulator using electrical energy at night or other. One thus obtains the supplementary heat needed under such circumstances.

It should be noted that battery BR, run through by the air expelled from the building in consideration, thus recovers a large part of the heat by consumption of the

negative kilogram calories in the chilled water circuit thus established, the system in consideration then only having to provide compensation for the losses, exception made for yields, which are inherent to the building itself and to other factors such as decreased visiting or others.

In another phase corresponding to reduced heat needs of the system, until approaching the natural average thermal equilibrium of the system, the consumption of negative kilogram calories of the chilled water circuit is strictly suited to the needs. In effect, thermostatic valve 11 modulates the level of passage of this water in battery or batteries BF, and energy regulator RE modulates the action of compressor Co.

In another phase corresponding to zero heat needs of the system and therefore in the case of its natural average thermal equilibrium point, thermal regulator RT causes the tipping of slide valve 15 in order to close way 15c and establish direct communication 15b, 15a, thus isolating battery BR, since slide valve 4 has not changed position.

The hot water produced in condenser C travels the circuit:

PC → C → BC → ways 4b, 4a (way 4c remaining closed) -> return to PC.

The chilled water produced in evaporator E travels the circuit:

PG → E → BF → ways 15b (open), way 15a, return to Pg.

The compressor, while ensuring the possible needs concerning consumption of negative kilogram calories of cold batteries BF, which are regulated by valve 11, is modulated to the minimum mechanical power by energy regulator RE. When this natural average thermal equilibrium of the system is reached, this corresponds to a slight excess of calories, and it is then possible to absorb these calories in a water heater such as A, which instead of being an external energy consumer, becomes a hot water producer, which can be used on site for sanitary needs or others.

The last phase to be considered here is that of the cooling needs of the system. In this phase, thermal regulator RT acts on slide valve 4 in order to connect ways 4a and 4c while closing way 4b. Slide valve 15 maintains the short circuit through hose 14 (way 15c closed, ways 15b and 15a in connection) while slide valve 4 establishes the long circuit for the hot water, thus causing it to pass through battery BR which then functions as air condenser, evacuating the excess of calories to the exterior, thus heating as it exits the air expelled from the building where the system in consideration is installed.

It should be stressed that battery BR could be arranged differently than on the path of travel of the air leaving the building and that it could at least partially be immersed in another medium, such as a water table with a high possibility of exchange and a quasi-constant temperature, for reasons of investment savings in particular, but in

that case with less recovery of the calories or the negative kilogram calories developed inside of the building.

In conclusion, the installation described has a single hydraulic loop, whose configuration can be modulated, giving rise to the advantages hereafter:

- continual transfer, from one place to another in a building provided with such an installation, of the consumed negative kilogram calories or calories: negative kilogram calories consumed at BF or BR, for example, lead to the appearance of a roughly equal number of calories available at BC, with increase due to the mechanical energy necessary for the transfer, hence reduced overall energy consumption;

- availability, in the summer as well as the winter (with no between-season discontinuity due to cycle reversal, as in the traditional installations) of all the potential heating or cooling power;

- complete recovery, until an exterior temperature of 46°C, of all the energy necessary for heating the fresh air supplying the building in consideration;

- recovery of the calories of the exterior air introduced as fresh air necessary for combating losses for all exterior temperatures higher than +6°C.

This limit of +6°C is set by the nature of the liquid coolant which is chosen to be water in the case of this numerical value, in order to avoid any icing or frost.

- simple regulation, using main regulator RT which acts on the circulation of the liquid coolant distributed in the various loops of the network; with possible supplementary addition of the thermostatic regulators of batteries and energy regulator RE given to compressor Co;

- possibility of completely isolating recovery battery BR from the condenser and evaporator exchangers especially in the case of use of a water table, which eliminates the risks of fouling of the first two exchangers;

- reduction of the wet bulb temperature of air intake in cooling cycle. Case in which exchanger BR is thus used by sending the air extracted from the building to this recovery exchanger in view of cooling the water of the circuit passing through condenser C, in "summer" operating conditions;

- possibility of providing hot water by extraction of the calories of the interior atmosphere of the building, for sanitary use in particular, in summer operating conditions.

It goes without saying that, without leaving the scope of the invention, it is possible to provided changes to the forms of execution just described.

Claims

1. A heat pump system which has a compressor with a liquid refrigerant associated with an evaporator and a condenser, respectively the source of cold and the source of heat in a liquid coolant network, characterized by the fact that said network has a number of loops, one being a short loop for heat passing through said condenser and at least one heating exchanger, the other being a short loop for cold passing through said evaporator and at least one cooling exchanger, these two loops being withdrawn [sic; and], downstream from their exchangers and upstream from the condensers of [misprint?, should possibly be et = and] evaporator by connections, between which a recovery line extends on which at least one corresponding exchanger is inserted, three-way slide valves being placed between the connection upstream from the condenser and from the evaporator and corresponding short loops.

2. A system according to Claim 1, characterized by the fact that the control of these three-way slide valves is ensured in a series, by a main regulator whose thermometric detector element is placed on the short loop for heat, upstream from the corresponding heating exchanger.

3. A system according to Claim 1 or Claim 2, characterized by the fact that a secondary energy regulator is provided with a thermometric detector element placed on the short loop for cold, upstream from the corresponding exchanger and whose control element determines the quantity of energy provided to said compressor.

4. A system according to any one of Claims 1 to 3, characterized by the fact that the exchangers of the loops for heat and cold are provided with a bypass whose opening is regulated by two-way valves actuated by thermostatic regulators, which are sensitive to the temperatures of intake of liquid coolant in said exchangers.

5. A system according to any one of Claims 1 to 4, characterized by the fact that said main regulator has two intakes, one of which is on said loop for heat and the other of which is sensitive to the exterior temperature.

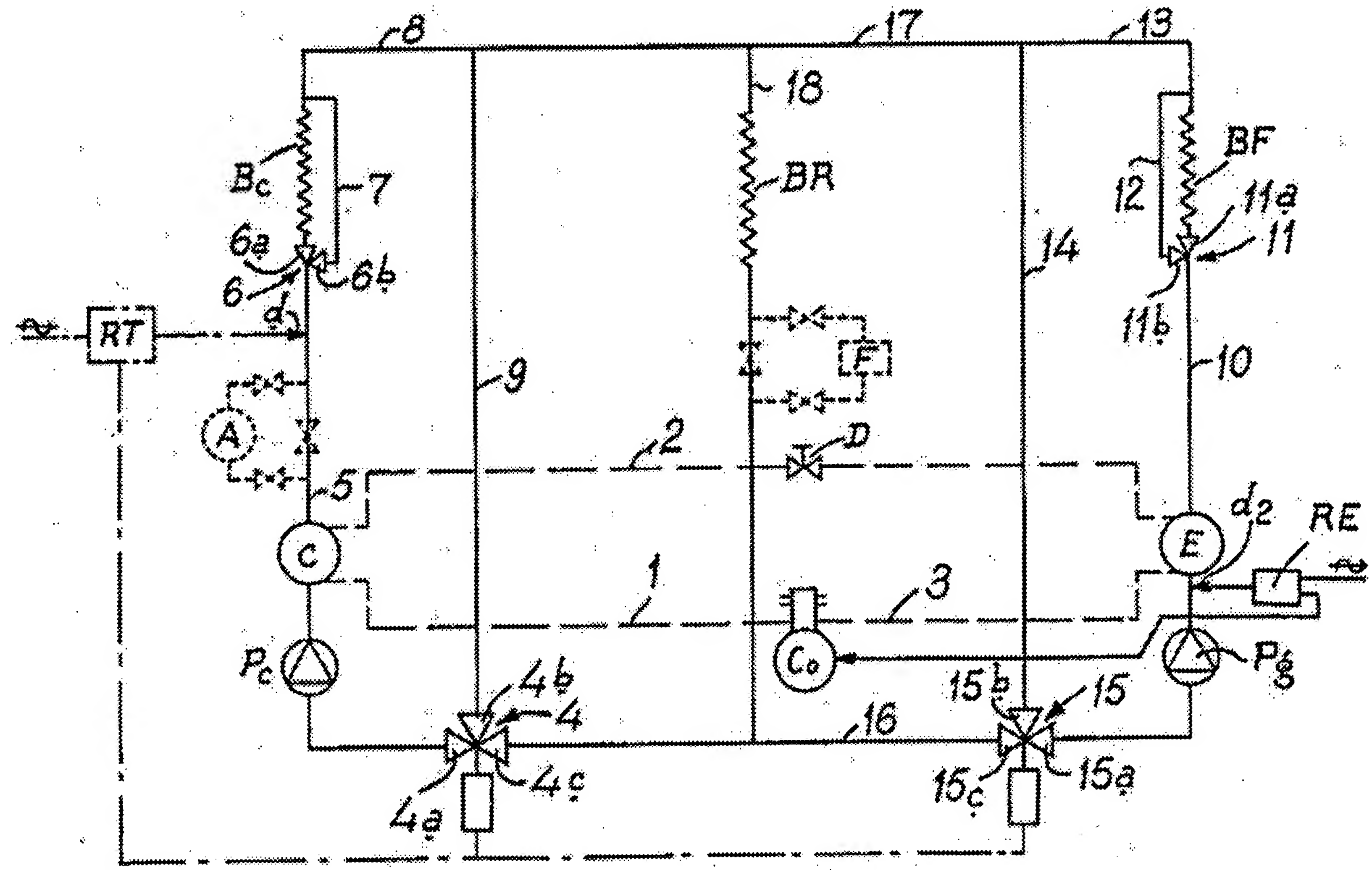
6. A system according to any one of Claims 1 to 5, characterized by the fact that each of the short loops has a circulatory pump for the corresponding liquid coolant, pumps which are situated upstream from the respective exchangers.

7. A system according to any one of Claims 1 to 6, characterized by the fact that the short loop for heat has, upstream from the corresponding exchanger, an accumulator, a generator, or both at the same time, capable of receiving additional exterior thermal energy.

8. A system according to any one of Claims 1 to 7, characterized by the fact that the recovery line moreover passes through an element forming an additional refrigerating source.

9. A system according to any one of Claims 1 to 8, characterized by the fact that the recovery exchanger can is exposed to the air extracted from the building.

10. A system according to any one of Claims 1 to 9, characterized by the fact that said recovery exchanger is exposed, at least in part, to a fluid with a high thermal capacity with a roughly constant annual temperature.



RT
8
17
13
18
Bc
7
BR
BF
12
11a
11
6a
6
6b
14
11b
d1
9
F
10
A
2
D
5
d2
RE
E
C
1
3
Co
Pc
Pg
4b
4
15b
15
16
4a
4c
15c
15a